THERMO-ECONOMIC ANALYSIS OF A FRENCH FRIES PROCESSING PLANT AT LAMBERT'S BAY

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DECLARATION

I, Johan Potgieter, submit this thesis in fulfilment of the requirements of the degree of Master of Science in Engineering. I claim that this is my original work and that it has not been submitted in this or in a similar form for a degree at any other university.

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ABSTRACT

In the literature study energy efficiency is discussed in general, as well as certain critical areas of importance to this study. In addition, measuring and monitoring equipment, and energy inefficiencies in steam and refrigeration systems are reviewed briefly.

In the energy analysis, an energy audit strategy is discussed in general. A walkthrough audit of the plant was conducted with specific focus on visible losses in the steam, refrigeration and production line systems. An energy analysis, as discussed in Chapter 3, indicates the main energy consumers, with steam being the biggest consumer of energy.

The main consumers of refrigeration energy are the cold stores, flow freezer and blast freezer. Energy consumption in the cold stores can be minimised mechanically, while refrigeration energy of the flow freezer and blast freezer can be minimised through the modification of production activity.

The main consumers of steam at the processing plant are the dryers, oil fryer, blanchers and steam peeler. Improved energy savings at the dryers can be obtained through optimisation of moisture and heat transfer mechanisms, while the energy of the blanchers and steam peeler can be combined by means of heat exchangers. The transfer of waste energy by means of a finned-tube heat exchanger from the steam peeler to the blanchers was investigated.

The newly installed coal boiler shows capacity for improving the quality of steam, as well as efficiency, by incorporating an economiser and separator for improving steam quality, automatic TDS control and blow-down heat recovery.

The product life cycle is discussed considering future automation that could lead to energy and labour savings.

Lastly the utilisation of product waste as a future research subject is discussed.

A confidentiality agreement was entered into with Oceana.

OPSOMMING

In die literatuurstudie word energie-effektiwiteit oor die algemeen bespreek, asook sekere kritieke areas wat vir die ondersoek van belang is. Hierbenewens word meettoerusting vlugtig bespreek, asook energie-oneffektiwiteite in die verkoeling- en stoomstelsels.

In die energie-ontleding word ’n energie-ouditstrategie in die breë bespreek. ’n Deurstap-oudit van die aanleg is uitgevoer met spesifieke fokus op sigbare verliese in die verkoelingstelsel, stoomstelsel en produksielyn. Die samestelling van energieverbruik in die aanleg word uiteengesit in Hoofstuk 3, waar dan ook aangedui word dat stoom die grootste energieverbruiker is.

Die hoofverbruikers van verkoelingsenergie is die koelkamers, die deurvloei-vrieskamer en die blitsvrieskamer. By die koelkamers kan verliese meganies geminimeer word, terwyl veranderinge aan produksie-aktiwiteite energieverbruik by die deurvloei-vrieskamer en die blitsvrieskamer kan verlaag.

Die hoofverbruikers van stoom by die verwerkingsaanleg is die droërs, oliebraaier, blansjeerders en stoomskiller. Energie-effektiwiteit by die droërs kan verhoog word deur vog- en warmte-oordrag optimaal te laat plaasvind deur korrekte instandhoudingsprosedures. Energieverbruik by die stoomskiller en blansjeerders kan deur middel van warmteruilers gekombineer word. ’n Ondersoek na die energie-integrasie van die stoomskiller en die blansjeerders is dan ook uitgevoer. Die pas geïnstalleerde steenkoolketel toon ruimte vir die verhoging van energie-effektiwiteit deur die daarstel van ’n ekonomiseerder – ’n skeier wat die gehalte van die stoom verbeter, outomatiese TDS-beheer en afblaasherwinning.

Die produk se lewensiklus word bespreek ten einde toekomstige outomatisering te motiveer in terme van energie- en arbeideffektiwiteit, asook die uitskakeling van onnodige blootstelling van die produk aan omgewingstemperature.

Laastens word die herwinning van afvalstowwe as ’n toekomstige navorsingsprojek bespreek.

’n Vertroulikheidsooreenkoms is met Oceana gesluit en word eerbiedig.

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A	Single payment in a series of n equal payments [R]
A_o	Outside area [m^2]
A_i	Inside area [m^2]
A_{fr}	Frontal area [m^2]
COP	Coefficient of Performance
COP_R	Coefficient of Performance for Refrigerator
COP_{RNH3}	Coefficient of Performance for Ammonia Refrigerator
c_{plugin}	Specific heat of dry air in [J/kgK]
$c_{pluguit}$	Specific heat of dry air out [J/kgK]
$c_{pvlugin}$	Specific heat of saturated water in [J/kgK]
$c_{pvluguit}$	Specific heat of saturated water out [J/kgK]
D	Diffusion coefficient [kg/ms]
D	Diameter [m]
e_f	Fin effectiveness
$\dot{E}_{steam,in}$	Energy flowrate of steam in [kW]
$\dot{E}_{steam,out}$	Energy flowrate of steam out [kW]
$\dot{E}_{product,in}$	Energy flowrate of product in [kW]
$\dot{E}_{product,out}$	Energy flowrate of product out [kW]
F	Future value [R]
F	Friction factor for smooth walls
G	Mass flux [kg/m^2s]
H	Specific enthalpy [kJ/kg]
h_a	Specific enthalpy of dry air [kJ/kg]
h_d	Mass transfer coefficient [kg/m^2s]
h_{evapin}	Specific enthalpy of evaporator working fluid in [kJ/kg]
$h_{evapout}$	Specific enthalpy of evaporator working fluid out [kJ/kg]
h_{fby70}	Saturation enthalpy of liquid water at 70 ^oC [kJ/kg]
$h_{g,16bar}$	Saturated specific enthalpy of steam at 16 bar [kJ/kg]
h_s	Specific enthalpy of steam [kJ/kg]
h_w	Specific enthalpy of water [kJ/kg]
H	Enthalpy flow [W]

HHV	Higher heating value [MJ/kg]
H_{SUIT}	Enthalpy of steam out [W]
H_{WIN}	Enthalpy of water in [W]
i	Interest rate per period
i_{gwo}	Latent heat of water vapour at a saturation temperature of 0 ^{o}C [J/kg]
K	Thermal conductivity coefficient [W/mK]
$Kalwaarde$	Calorific value of heavy fuel oil [J/L]
L	Length [m]
$\dot{m}_f$	Mass flowrate of saturated liquid [kg/s]
$\dot{m}_g$	Mass flowrate of saturated gas [kg/s]
m_{HFO}	Mass flow of heavy fuel oil [kg/hr]
m_{lugin}	Mass flow of air in [kg/s]
$\dot{m}_{potato}$	Mass flowrate of potatoes [kg/s]
$\dot{m}_{product,in}$	Mass flowrate of potatoes in [kg/s]
$\dot{m}_{product,out}$	Mass flowrate of potatoes out [kg/s]
$\dot{m}_{steam,in}$	Mass flowrate of steam in [kg/s]
$\dot{m}_{loss,steam}$	Mass flowrate of steam lost for further heat recovery [kg/s]
$m_{uitlaat}$	Mass flow of combustion products [kg/s]
m_w	Mass flow of water in/steam out [kg/s]
$\dot{m}_{waste}$	Mass flowrate of potato waste [kg/s]
M_a	Molar mass of air [kg/kmol]
M_v	Molar mass of vapour [kg/kmol]
n	Number of interest period
NCV	Nett calorific value [MJ/kg]
P	Present value [R]
P	Pressure [Pa]
P_{atm}	Atmospheric pressure (101 300 Pa)
$Power_{ideal}$	Ideal power [kW]
$Power_{real}$	Real power [kW]
P_{sat70}	Saturation pressure at 70 ^{o}C [Pa]
Pr	Prandtl number
Q_E	Heat flow of exhaust gases [W]

Q_H	Heat flow from/to warm source/sink [W]
Q_L	Heat flow from/to cold sink/source [W]
Q_O	Heat losses to environment [W]
Q_V	Heat contents of combustion product [W]
Q_W	Heat flow into water [W]
R_{lug}	Specific gas constant of air [J/kgK]
r	Radius [m]
r_i	Inside radius [m]
r_o	Outside radius [m]
Sh	Sherwood number
Sc	Schmidt number
T	Temperature [K]
T_H	Temperature at warm source/sink [K]
T_L	Temperature at cold sink/source [K]
T_{lugin}	Temperature of air in [K]
T_{luguit}	Temperature of air out [K]
U	Overall heat transfer coefficient [W/m^{2}K]
U_{design}	Overall design heat transfer coefficient [W/m^{2}K]
U_{actual}	Overall actual heat transfer coefficient [W/m^{2}K]
V_{HFO}	Volume flow of heavy fuel oil [L/hr]
V_a	Molecular volume of air [m^3/kmol]
$\dot{V}_{loss,potato}$	Volume flowrate of lost potatoes [m^3/s]
V_v	Molecular volume of vapour [m^3/kmol]
w_a	Specific work actual [W/kg]
W_{pump}	Specific work pump [W/kg]
w_s	Specific work shaft [W/kg]
$W_{net,in}$	Specific work net in [W/kg]
W	Work [W]
W_{PUMP}	Pump work [W]
$Yield_{alc}$	Alcohol yield [l/ton]

Greek Symbols

Δx	Length of segment [m]

η	Efficiency
η_C	Carnot efficiency
η_{con}	Conversion efficiency
$\eta_{Plant/System}$	Efficiency factor of plant/system
η_{utils}	Utilisation factor
ρ	Density [kg/m^3]
ρ_{potato}	Density of potatoes [kg/m^3]
ρ_s	Density of steam [kg/m^3]
ρ_w	Density of water [kg/m^3]
μ	Dynamic viscosity [Ns/m^2]
ν	Specific volume [kg/m^3]
$v_{g,16bar}$	Specific volume of saturated steam at 16 bar [kg/m^3]
υ	Kinematic viscosity [m^2/s]
v_{fby70}	Specific volume of saturated liquid at 70 ^{o}C
Φ	Relative humidity [kg water vapour/kg water vapour at saturation]
ω	Specific humidity [kg vapour/kg dry air]

1 INTRODUCTION

1.1 Energy Efficiency as National Interest

A thermo-economic analysis is an efficiency analysis of the application of thermal energy in a plant. In this document, an approach towards an energy efficiency analysis for a French fries processing company at Lambert's Bay, South Africa, is developed.

Energy efficiency is often a neglected income for companies, especially in South Africa, where energy costs are of the cheapest in the world and the interest rates are high, which increase payback time. Many people consider the energy efficiency industry to be a billion-Rand industry, accompanied by many advantages. It was found that safety and production improve when energy is used more efficiently.

Ghana is already beginning to reap the fruits of its energy efficiency programme, which started in November 1997. The Executive Director of the independent Ghana Energy Foundation, Dr Alfred Ofosu-Ahenkorah, states that the primary weapon used to promote the efficient use of energy is education – making people aware of simple housekeeping measures – and the second weapon is technology, which, of course, costs money (Zhuwakinyu, 2001). All of the above strengthens the need for an energy efficiency analysis.

1.2 Energy Efficiency in the French Fries Processing Industry

Most French fries processing plants use the same methods of processing and, thus, more or less the same energy transport methods. Steam and refrigeration systems are the largest energy consumers in the French fries industry, therefore the steam and refrigeration system will be the main focal point in this document.

2 LITERATURE STUDY

2.1 Energy Efficiency

Energy is one of the largest controllable costs in most organisations and companies. There is considerable scope for reducing energy consumption and, therefore, costs. A company should realise that good housekeeping results in more efficient use of energy, and that company staff need to be educated before capital is spent on equipment that will improve energy efficiency.

2.1.1 The Energy Efficiency Earnings Strategy

A general energy efficiency earnings strategy was compiled by the Energy Research Institute (ERI) of the University of Cape Town (The 3E Strategy, How to Save Energy and Money, 2000) to enable companies to determine the energy efficiencies of their energy consumers. This strategy was modified for the special application required for the French fries processing industry and, to a certain extent, is employed in this thesis. According to the ERI, the five fundamental aspects of an energy management strategy are:

i. The need for a company energy efficiency strategy or plan and the basic outline for a cost reduction program.
ii. Purchase and cost control and a consumption audit of primary energy usage.
iii. Framework and methodology for monitoring and targeting energy savings.
iv. Savings in energy usage through positive practical methods for improving the efficiency of plant and industrial processes.
v. Financial appraisal of energy efficiency.

The following discussion will explain how this energy efficiency earnings strategy should be implemented. According to the ERI, energy savings projects may be divided into four categories:

1. Improved housekeeping (maintenance, etc.)
2. Low-cost modifications and improvements
3. Refitting existing systems with new parts and equipment
4. Major capital expenditure

An energy audit of a facility is conducted by means of a walkthrough audit, a diagnostic audit and the identification of potential energy saving areas.

The walkthrough audit is a tour through the facility or plant, looking for obvious signs of energy wastage. This audit is more significant when conducted by a person who is familiar with process insulation and energy management. Typical signs of energy wastage that can be noticed during a walkthrough audit include missing or damaged insulation, hot or cold surfaces, wet insulation, deteriorating insulation coverings or protective finishes, missing or damaged vapour retarders, gaps in insulation at expansion or contracting joints, steam leaks or visible steam wastages.

As soon as these signs are identified in the walkthrough audit, a diagnostic audit is required in order to determine the existing energy loss, the reduction in energy loss which would result at the expense of modifications done or capital spent. Simple payback calculations are done to determine the financial viability of the opportunity.

In order to identify potential areas of energy saving, the first step is to establish the quantity and cost of the energy and utilities used on site. The typical energy and utilities are fuel, oil, coal, gas, electricity, water and, in some instances, vehicle fuel. Information regarding quantity and cost can be obtained through accounts of the past year.

On completion of the audit, it is essential to investigate whether the utilities are being purchased competitively. Management control is an essential element. Savings are often made available by managing resources more effectively through the use of standard monitoring and targeting techniques. The latter is a disciplined approach to energy management, which ensures that energy resources are used to the maximum, as well as the monitoring of savings brought about by improved purchasing (better prices for energy resources) and through energy-saving investments.

An accurate picture of recent energy consumption and costs is needed in order to make decisions for energy efficiency improvements. To obtain information of recent energy consumption and costs, the following sources can be used: at least one year's utility invoices for fuel, electricity and water, site energy records, and sub-metering and production information.

A table, illustrating consumption and costs, must be drawn up. The monthly trends in consumption must be plotted correspondingly. In this way, variation during the year can be seen and the trend examined to determine any untoward pattern of consumption.

The energy consumption is divided into the main services, processes and end users and drawn into a Sankey diagram, showing the percentage breakdown of each energy type. With the Sankey diagram, an informative picture of energy flow in the plant is generated. Figure 1 illustrates a Sankey diagram for a small factory.

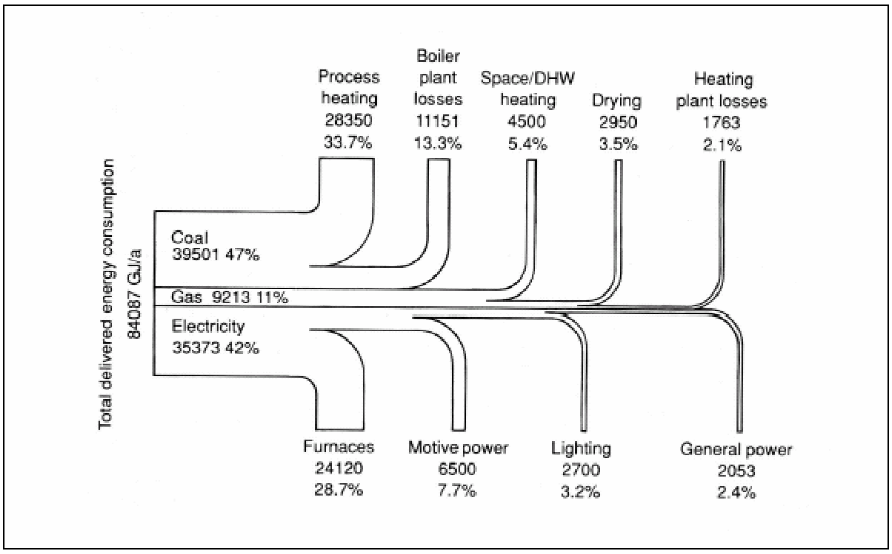

Figure 1: Sankey diagram for a small factory, indicating the energy flow (Energy Research Institute, 2000)

However, it is not always possible to have all the necessary information at hand to draw a Sankey diagram. More detailed information can be obtained by studying the seasonal variations of energy consumption through the use of instrumentation.

Obtaining the best energy price depends on market knowledge and negotiation skills. In this case, the clients have their own in-house expertise, although it is necessary to confirm the competitiveness of the tariffs.

2.2 Energy Efficiency Analysis of Food Processing Facility

Most French fries processing facilities use process steam, refrigeration, compressed air and electric motors in their operations. Process steam and refrigeration are the biggest energy consumers in the French fries industry and are therefore the focus of this thesis.

What about involving management in an energy efficiency analysis? Involving management in an energy efficiency analysis is essential for its successful implementation, because energy efficiency is not only the physical implementation of certain equipment and procedures, but also a change of philosophy that a company and its personnel needs to adopt. The overall effect of this mindset also leads to improved housekeeping and maintenance, as well as increased production and utilisation of production machines.

2.2.1 Walkthrough Audit

In order to determine whether an energy analysis will be financially viable for the company, it is necessary that an approximate energy audit, called a walkthrough audit, be conducted. In the walkthrough audit different systems, subsystems and components need to be identified and obvious wastages highlighted. At most French fries processing facilities significant energy usage occurs with process steam, refrigeration, compressed air and electric motors.

In the walkthrough audit, visually detectable wastages are process steam and refrigeration wastages. Compressed air inefficiencies are mostly due to air leaks, which can be sensed by hand. Electric motor inefficiencies unfortunately need to be measured and cannot be detected or calculated during the walkthrough audit.

2.2.2 Diagnostic Audit

To determine the approximate energy losses by means of the walkthrough audit, a diagnostic audit is necessary. The diagnostic audit uses approximate formulas to determine energy losses and simple payback calculations are done to determine the financial feasibility of further detailed analyses. Accounts of electricity, water, fuel and other consumables need to be analysed in terms of the production rate and seasonal variations for the presence of trends.

2.2.3 General Energy Audit

With the general energy audit, the energy usage rate of the plant/system for a certain production rate is determined and compared against the ideal energy usage rate for that production rate in order to calculate the plant/sytem efficiency. For electrical systems, this can be done by measuring the electric current of the main electrical supply and multiplying it by the voltage difference. For steam systems, a steam flow meter can be installed, but it is quite expensive. To determine a plant or system's ideal energy usage rate, the design documentations could be used to determine ideal energy usage rates. The plant/system efficiency is defined by

$$\eta_{plant/system} = \frac{Power_{ideal}}{Power_{real}} \qquad (2.\ 1)$$

where $Power_{ideal}$ is the power that would have been consumed for an ideal system with no losses and $Power_{real}$ is the power consumed for a real system with losses. Identifying losses that contribute to the inefficiencies, estimating the cost of modifications and/or improvements to decrease the inefficiencies and calculating the payback period, will provide an indication of the feasibility of an energy analysis. A general plant energy efficiency will not always be possible to calculate, but by improving the efficiency of subsystems, it is likely that the overall efficiency will improve.

Energy Audit for Subsystems

After the general energy audit and if the calculations indicate possible savings, energy audits are done on each of the subsystems. According to Blanchard and Fabrycky (1998), a system is an assemblage or combination of elements or parts forming a complex or unitary whole. In this case, the French fries processing facility is the system with a combination of subsystems forming a unitary whole on their own as elements or parts in the system. With this definition in mind, the following subsystems are significant energy consumers at a typical French fries processing facility:

i. Steam system
ii. Refrigeration system
iii. Electricity system
iv. Compressed air system

An energy audit for a system comprises an efficiency analysis, which measures the real energy usage rate (power usage) for a certain production rate and compares it against the ideal energy usage rate. This efficiency analysis will then be carried down to subsystem level and, eventually, to a component level where the causes of energy efficiencies must be identified. Thus, the efficiency of a system is calculated by:

$$\eta_{system} = \frac{Power_{ideal}}{Power_{real}} \qquad (2.\ 2)$$

Energy Audit for Components of a Subsystem

If the plant or system efficiency is low in comparison with other plants or systems, an energy audit on the components of the subsystem can be done in order to identify the inefficient subsystem.

Suitable meters can be installed to determine the contribution of each system/component to the total plant/system power input.

Typical systems and their components for steam and refrigeration systems are discussed below, as well as how an energy audit should be approached. Checklists need to be compiled for quick identification of possible areas of energy wastage. To conduct an energy audit, the incoming and outgoing energy rates need to be documented and an energy flow analysis needs to be conducted. The correct measuring equipment must be used for successful measurements and therefore knowledge of available measuring equipment is necessary.

2.2.4 Energy Audit of Steam

Steam is used for heating and process work, as it is an ideal carrier of heat. The three main advantages of steam are: (1) steam transfers heat at a constant temperature, (2) the temperature of steam is dependent on the steam pressure, which results in a simple method of temperature control and (3) steam is compact in terms of heat content per unit volume, which means that heat can be conveyed in simple piping systems.

The efficiency of steam consumption is not as easily measured as the thermal efficiency of a boiler or refrigeration system, and as a result, it is frequently neglected. Steam consumption includes process steam from the point of generation, distribution, end-use to the recovery of condensate return. Few companies know what their steam costs are, yet this should form an important part in the costing of any product. There are various ways of increasing the efficiency of a plant's steam consumption. A poorly operated and maintained system can use twice the amount of fuel that a well-operated and maintained system will use. (http://www.energywise.co.nz/content/pdf/EECA.STEAM2.pdf)

Any steam system can be divided into the following subsystems:

i. Steam generation
ii. Steam distribution
iii. Steam end-use
iv. Condensate recovery

Steam Generation

Steam is generated in a boiler and, depending on the process temperatures and amount of steam to be utilised, the size of the boiler is determined. Various forms of energy can be used to produce steam, but in this document the focus will be on fossil fuel boilers, specifically coal and heavy-fuel oil boilers. Fuel is burned in a burner (in the case of heavy-fuel oil) or in a combustion chamber (in the case of coal) to produce steam in the boiler, which is then transported to its different destinations within the facility. To optimise the operation of boiler plants, it is necessary to understand where energy wastage is likely to occur. The figure below indicates the general inputs and outputs for a typical oil fuel-type boiler (the outputs of a coal-fired boiler will differ slightly in value).

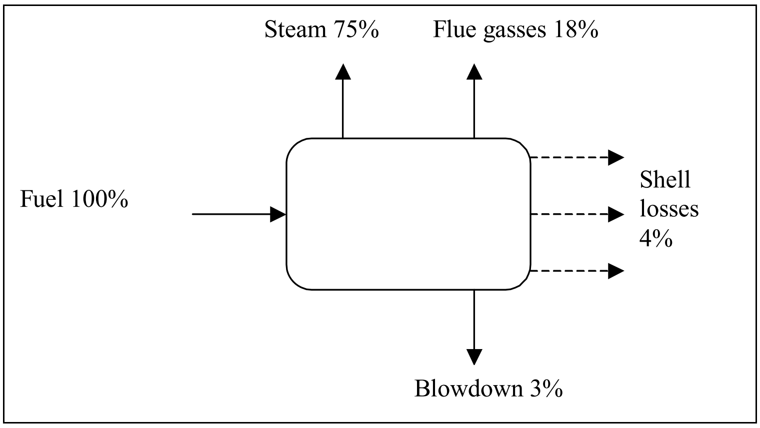

Figure 2: Boiler inputs and losses

(Energy Research Institute, 2000)

Efficiency can be increased through proper maintenance, blow-down heat loss minimisation, excess air reduction, flue gas heat recovery and combustion air preheating.

Significant energy savings can be obtained through proper maintenance, as set out below:

i. Maintain proper burner adjustments. The operator must be able to identify the appearance of a proper burner flame. The flame should be checked frequently and always after any significant change in operating conditions.

ii. Overhaul regenerative air heater seals. Excessive air could leak from the air side to the gas side of the air heater if the seals are in poor condition, which could result in increased forced draft fan power consumption and may reduce the maximum boiler capacity.

iii. Check boiler easing for hot spots. Hot spots are indications of excessive heat losses from the boiler enclosure. The temperature of the outer skin of the boiler should not exceed 50 ^{o}C, although higher temperatures may be inevitable where insulation cannot be installed.

iv. Replace or repair missing and damaged insulation.

v. Replace boiler doors and repair leaking door seals.

vi. Repair malfunction steam traps. Steam traps may fail in the open or shut position. An open steam trap will pass excessive quantities of steam and increase the heat loss. A closed trap will not allow condensate to escape. If the closed trap is connected to a heat exchanger, the heat exchanger will gradually be filled with condensate and eventually fail to operate. If the heat exchanger is outside, the condensate could freeze in winter and damage the tubes of the unit. If the closed steam trap is draining a steam line, excessive condensate may build up in the line and cause water hammer in the system. This could damage fittings and equipment. A regular steam trap maintenance programme will help minimising energy losses caused by the abovementioned failures.

vii. Calibrate and tune measurement and control equipment. A common cause of deteriorating boiler efficiency is operating at higher excess air values than necessary.

Blow-down heat loss depends on a variety of factors and typically varies between 1% and 6%. Factors contributing to blow-down heat losses are the total dissolved solids (TDS) allowable in the blow-down water, the quality of the makeup water (which depends mainly on the type of water treatment installed), the amount of uncontaminated condensate returned to the boiler house and boiler load variations. Correct checking and maintenance of feed water and boiler water quality, maximising condensate return, as well as smoothing load swings, will minimise the loss. The installation of blow-down heat recovery systems will also help to minimise the loss.

It is possible to calculate the exact amount of air that is needed for combustion for every fuel. In practice some surplus or excess air is required to ensure complete combustion, the amount varying with the type of fuel being burned. Any further excess air that is heated passes through the boiler and out of the stack, thereby reducing system efficiency. Simply adjusting the excess air is not necessarily sufficient. The air must mix with the fuel at the correct point. Almost all combustion systems use two sources of combustion air: the air which immediately mixes with the fuel to initiate combustion (primary air) and that used to complete the combustion (secondary air). In order to obtain complete clean combustion it is essential that these two sources of air are available in their correct ratio. Automatic controls could be added to a boiler system to ensure correct air ratios.

Most heat losses in the boiler are in the flue gas. The flue gas temperature should be as low as possible above the dew point of sulphur gases, which could otherwise condense into acids, attacking the stack and associated equipment. Flue gas economisers have been in use for a long time on both shell and water tube boilers of older design. Introducing an economiser to the boiler breeching will increase the pressure drop in the flue gas system. Feed-water piping modifications, economiser support and possible breeching modifications must be evaluated.

Air preheaters are large and overall less efficient than economisers. In order to improve thermal efficiency by 1% the combustion air typically needs to be raised by 20 ^{o}C. The usual sources for combustion air preheating include the heat remaining in the flue gases, higher temperature air drawn from the top of the boiler house and heat recovered by drawing air over or through the boiler casing to reduce shell losses.

Steam Distribution

The steam distribution system consists of pipes and steam fittings that distribute the steam from the generation plant to various points of use. Unlagged piping, valves and leaks in the steam system contribute to losses. Insulating unlagged sections of pipe work and fittings is one of the simplest and most cost-effective ways of increasing the energy efficiency of a heat distribution system.

Steam should be generated at the lowest possible pressure necessary to meet the maximum temperature required by the equipment in the system. There are, however, certain benefits of distributing steam at high pressure. High-pressure steam distribution minimises the size of the piping required, thereby reducing capital costs. This, of course, is financially viable only in the design phase for new plants where there is no existing piping. High-pressure steam distribution also minimises the amount of insulation material required in small-diameter pipes. In larger pipes, high pressure increases the recommended insulation thickness and therefore the benefits of less insulation are not always achieved.

Steam that is reduced in pressure through the use of a pressure reduction valve (PRV) may have to be de-superheated before being used in a process. When the pressure of a volume of saturated steam is reduced, the heat content is not lost. The excess heat above that which the saturated steam at the new pressure can hold, turns into sensible heat in the steam, raising its temperature. In cases where maximum temperature is a critical process parameter, this excess heat must be removed at a point where, often, no other user is available. The heat is therefore lost to the system, reducing its overall efficiency.

Once the necessary system pressure has been determined, the pipes must be sized correctly. If the pipe is too small, insufficient steam at a high enough pressure will get through to the process. Too large a pipe simply means that surface heat losses are increased or more insulation is required. Either way the system efficiency will drop. The number of bends and valves affects the pressure drop along a given length of pipe. A study of unnecessary bends in the piping and whether it would be worthwhile in the long run to replace it with piping that contains less bends and valves can also be done.

As steam cools down, it reverts to water. Condensate in a steam line is inevitable, but it can be potentially disastrous. Condensate laying at the bottom of a pipe effectively reduces the pipe's cross-sectional area, thereby requiring increased velocities and causing a bigger pressure drop. At worst, the layer of condensate becomes deep enough to be picked up by the steam, forcing it as a bullet or plug down the pipe. In extreme cases, this water hammer can lead to sudden failure of the pipe or fittings, such as valves.

If the condensate is carried forward into the process machines, again the possibility of damage arises. Wet steam builds up a thick film on heat transfer surfaces, reducing the overall effectiveness of the process machines. A good layout of steam pipe-work will ensure provision for the removal of condensate from the distribution system before it can cause a problem.

The heat transfer rate from condensing steam to a surface is very high, but can be seriously impeded by films of air or water. Appropriate air and steam condensate removal techniques will improve the overall efficiency of heat transfer. Air accumulates in all steam spaces when steam supplies are turned off and systems are allowed to cool down. When steam supplies are turned on, the air has no option but to mix with the steam unless it is allowed to escape from the system. The removal of air can be carried out by either manual or automatic air venting.

The purpose of a steam trap is to remove condensate and air and to retain steam. There are three main categories of steam traps, namely mechanical, thermostatic and thermodynamic steam traps. To determine which steam trap to use, it is useful to match the characteristics of the trap to the heat requirement for the equipment in question.

Condensate in the piping system must be removed promptly to avoid water hammer. Ensure that drip legs are placed at the right places for the removal of condensate. Drip legs must be provided at all low points and at natural drainage points in the system. Natural drainage points include the end of main steam pipes, bottoms of risers and before pressure regulators, control valves, insulation valves, pipe bends and expansion joints.

Separators are designed to remove entrained water particles from the steam so that they can be drained away as condensate through a conventional steam trap. The separator also helps to prevent contaminants being carried further into the steam system. Separators are commonly fitted just after the boiler crown valve and before major steam-using equipment.

A steam separator is an in-line device used to remove entrained condensate particles. These condensate particles can originate from boiler carryover, or they could be the unwanted result of undersized pipe and equipment. A well-designed steam system minimises the accumulation of condensate within the system and immediately carries off any accumulating condensate. With a properly sized and configured piping system, well-located drip legs and correctly sized equipment a steam separator is unnecessary.

However, when a plant experiences reduced heat exchanger efficiency, erosion at pipe directional changes, erosion on in-line equipment and water hammer, the installation of a separator should be considered. These are all possible indicators of the presence of entrained condensate particles and the accumulation of condensate in the flow of steam. Unless a system is poorly designed, these symptoms are usually the result of badly planned, multiple expansions and modifications to a steam system.

If a system is expanded to and operated at its capacity any weakness in the design and/or installation will start to show. One of the first telltale signs that a steam system is at its capacity and has a weakness in its design is the above indication of condensate in the system. In order to alleviate the problem without going through the time and expense of resizing and replacing existing piping, a separator could be installed. A strategically placed separator cannot cure the problem, but it can prevent erosion and other damage caused to piping and in-line equipment due to the formation and build-up of condensate.

The steam separator is designed to work in-line and should be expected to remove approximately 95% – 98% of the entrained condensate particles. With various designs of the same theme, the separator functions as an enlarged section in a pipeline. Internally, there are baffle plates designed to either impinge the particles or force them to the outer perimeters of the housing or both.

As the steam (and the entrained condensate particles) enters the separator chamber it suddenly and momentarily looses some of its velocity due to the sudden enlargement of the chamber. The mass of the condensate particles propels them forward into the impingement baffles. In the case of a cyclone-type design, vanes or fins will force the particles into a high-velocity rotation inside the chamber. The particles will then be forced to accumulate either on the impingement baffle or the outer perimeter wall of the separator chamber and collect at a low point in the separator. Connected to the low point of the separator is piping that leads to a steam trap where the condensate will then be returned to the feed-water tank.

The need for a separator, aside from installing one to safeguard the system from possible boiler carryover, indicates a system problem. The separator is installed to control certain aspects of the problem and not to correct it. Depending on the severity and magnitude of the problem, it might be more cost-effective to research the problem itself, determine the cause and correct it.

Having gone through a lot of expenses in generating steam and installing a distribution system one needs to deliver the steam to the various steam-using equipment in the plant as efficiently as possible. 'Efficiency' means supplying steam to the users with a minimal loss in latent energy at a reasonable cost. This is where the steam trap comes into the picture. Without steam traps unabated condensate would form in distribution piping, creating a wide range of problems. In addition, there would be no essential control at the users. Steam would enter a set of tubes or a coil at the one side and come out at the other side as either steam, condensate or a two-phase mixture of the two, which is very dangerous, damaging and wasteful.

By installing steam traps in strategic locations throughout the distribution system, these problems can be alleviated. A steam trap on the outlet side of a heat exchanger allows the steam to reside there until all the latent heat energy is transferred and the accumulated condensate is carried off. With proper placement and specification of steam traps for these purposes one can create and maintain an efficient, cost-effective steam supply and distribution system. In knowing this, the next step is to determine the best trap to use for a given application. The following section discusses the various types of traps and the conditions for which each steam trap is suited.

Steam End-use

With steam utilisation the losses are split between heat loss in the system (leaks, et cetera) and heat loss caused by inefficient use at the steam utilisation point. A typical example of heat loss due to inefficient use is steam discharge heat recovery from the steam peeler in the French fries industry (Dutch National Team, 2000), where the heat content of the discharge steam is used to heat up the water at the preheater. One of the obstacles in utilising the steam are potato particles and potato peels in the discharge steam. These particles and peels can block a heat exchanger if the spaces between the fins or tubes are too small. A possible solution can be obtained by installing a duct or pipe over the steam peeler's discharge line, decreasing the speed of the discharge steam and causing the heavier particles in the steam to separate. This action makes the steam more useful.

Steam is used to convert heat into another form, such as hot air for drying or boiling water to cook a product. Processes that include the use of steam are blanching, drying and frying. To heat the water for blanching, the steam expands directly in the water, where the hot water is used to make the potatoes more elastic for the cutting of fries. The drying of the fries by removing moisture takes place in dryers, where steam heats up outside air to lower the relative humidity of the air in order to increase the rate of moisture removal through evaporation and capillary action. A shell-tube heat exchanger is used to transfer heat from the steam to the oil needed for frying the fries.

Steam Recovery

After giving up its latent heat to the process, steam condenses. The steam condensate carries a sizable proportion of the original energy content put in by the fuel. Both water treatment and blow-down costs can be reduced considerably through condensate recapturing, because condensate that has not been in contact with the process is chemically pure and therefore needs little water treatment apart from pH adjustments.

Possible Savings

Through proper maintenance, steam leaks can be detected at an early stage and insulation must be kept in a good condition. Undersized or incorrectly selected steam traps can result in poor heat exchanger performance and inefficiencies. Heat exchanger surfaces should be cleaned regularly of product build-up. Air should be removed by a combination of correctly fitted air vents and steam traps, which will allow air to pass. Uninsulated valves and flanges result in substantial losses, but insulation blankets, which can easily clip onto the valve, are available. Steam traps are essential to a steam distribution system, but leaks could occur. The traps can be tested for leaks by conventional methods such as checking sight glasses downstream of traps or listening for the noise of steam passing through the trap. Another sign of leakage could be excessive steam venting from a condensate receiver.

Measuring Performance

The boiler efficiency can be checked through metering the fuel used and steam produced. This is compared to the design efficiency of the boiler. At the process equipment the steam in or condensate return is metered and the amount of steam that the equipment used is then known. With the current production rate available, it can be compared to the design value of the equipment under investigation.

2.2.5 Energy Audit of Refrigeration

Most refrigeration systems are driven by a machine, which compresses and pumps refrigerant vapour and liquid in a closed circuit. Heat is absorbed and rejected by heat exchangers. These systems work according to a vapour compression cycle. The refrigeration system at Lambert's Bay Foods consists of an ammonia refrigerant cycle and therefore this section will discuss a typical vapour compression, ammonia refrigeration cycle.

In a refrigeration cycle heat must flow from a cold region (cold stores and freezers) to a warm region (ambient air). This is achieved by using a refrigerant, which absorbs heat at a low pressure and then boils or evaporates to form a gas. The refrigerant gas is then compressed to a higher pressure, hence a higher temperature, such that it transfers the heat it has gained to the environment (the ambient air) or water, where the refrigerant condenses. In this way heat is removed from a low-temperature source to a high-temperature sink through work input. The operation of a simple refrigeration system is illustrated in the T–s diagram in figure 3.

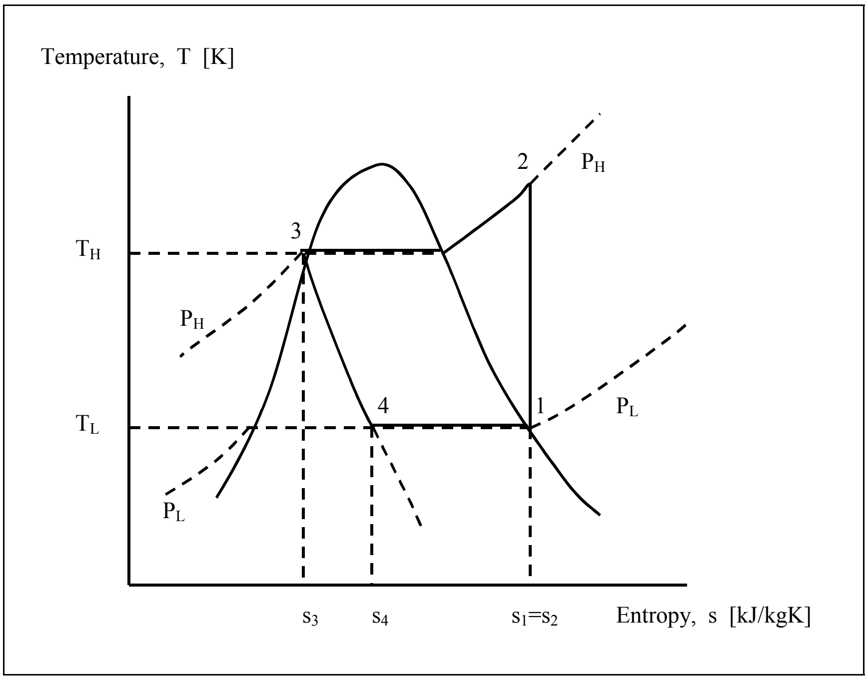

Figure 3: Typical vapour compression cycle

The refrigeration cycle can be divided into four processes (see figure 3). Firstly, low-pressure liquid refrigerant in the evaporator absorbs heat from its surroundings, usually air, water or another process liquid (processes 4 to 1). During this process it evaporates and is slightly superheated at the evaporator exit. Secondly, the superheated vapour enters the compressor where its pressure is increased (processes 1 to 2). A large increase in temperature will also occur, because a portion of the energy put into the compression process is transferred to the refrigerant. Thirdly, the high-pressure, superheated gas passes from the compressor into the condenser (processes 2 to 3). Initially, the cooling process de-superheats the gas before it condenses into liquid. The cooling for this process is usually achieved by using air or water. A further decrease in temperature occurs in the pipe-work and liquid receiver so that the refrigerant liquid is sub-cooled (cooled below its boiling point). Finally, the high-pressure, sub-cooled liquid passes through the expansion device, which reduces its pressure and controls the flow to the evaporator.

In order to determine the efficiency of the refrigeration cycle, an ideal refrigeration cycle must be used. A theoretical model, the Carnot cycle, represents the basic processes of a reversible heat engine (producing work from heat). The Reverse Carnot cycle transfers heat through the addition of work. Both these cycles are reversible, hence operating at maximum efficiency between two different environments. With the Reverse Carnot cycle the maximum theoretical performance can be calculated, thereby establishing criteria to which real refrigeration cycles can be compared. Although the Reverse Carnot cycle is a useful model to assist in the understanding of the refrigeration process, it has certain limitations. One limitation is the lack of accounting for changes of states. Figure 3 (see page 17) shows a vapour compression cycle, approximating the effect of the cycle on the refrigerant, and assuming ideal equipment.

The refrigeration effect is represented by the difference between the exit and entrance enthalpy of the evaporator. The coefficient of performance (*COP*) is obtained by dividing the refrigeration effect (Q_L) by the net power input ($W_{net,in}$).

$$COP = \frac{Q_L}{W_{net,in}} \tag{2. 3}$$

Refrigeration cycles are more complex than the theoretical vapour compression cycle discussed. Practical limitations, such as equipment size, system pressure and design temperatures at the evaporator and condenser, reduce the effectiveness of actual systems.

The performance of a refrigerator is expressed in terms of its coefficient of performance, which is defined as:

$$COP_R = \frac{desired\_output}{required\_input} = \frac{cooling\_effect}{work\_input} = \frac{Q_L}{W_{net,in}} \tag{2. 4}$$

Note that the coefficient of performance (COP_R) is generally bigger than 1. The Reverse Carnot cycle is the perfect refrigeration cycle and no irreversibilities are generated in the cycle. This cycle is a totally reversible cycle which consists of two reversible isothermal and two isentropic processes. It offers the maximum thermal efficiency for given temperature limits and serves as a standard to which actual power cycles can be compared. Since it is a reversible process, all four processes comprising the Reverse Carnot cycle can be reversed. In order to determine how closely the refrigeration cycle is performing to the ideal performance, a reference is needed – that reference is the Reverse Carnot cycle.

The coefficient of performance for a Reverse Carnot cycle can be determined if the temperatures between which the refrigerator works (i.e. the temperature of the heat source and the tempearture of the heat sink) are known. Theoretically, this is the best performance that a heat pump or refrigeration cycle can obtain between two temperatures, and the performance of any refrigerator working between two temperatures must be determined relative to this. The coefficient of performance for a Carnot refrigerator is as follows:

$$COP_R = \frac{1}{T_H / T_L - 1} \tag{2. 5}$$

The coefficient of performance increases as the difference between the two temperatures decreases. Shown in figures 4 and 5, are schematics of simple vapour-compression refrigeration cycles.

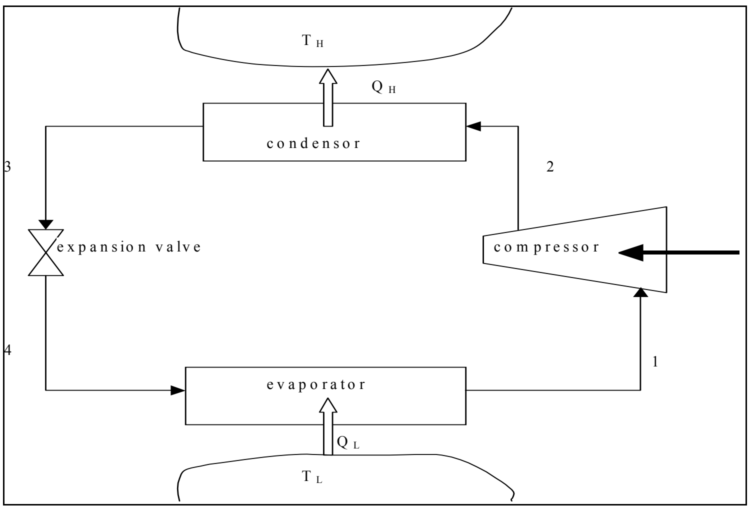

Figure 4: Simple vapour-compression cycle

The refrigeration cycle used at Lambert's Bay is similar to the simple vapour compression cycle, except for the addition of a receiver and flash chamber. The following schematic representation demonstrates this:

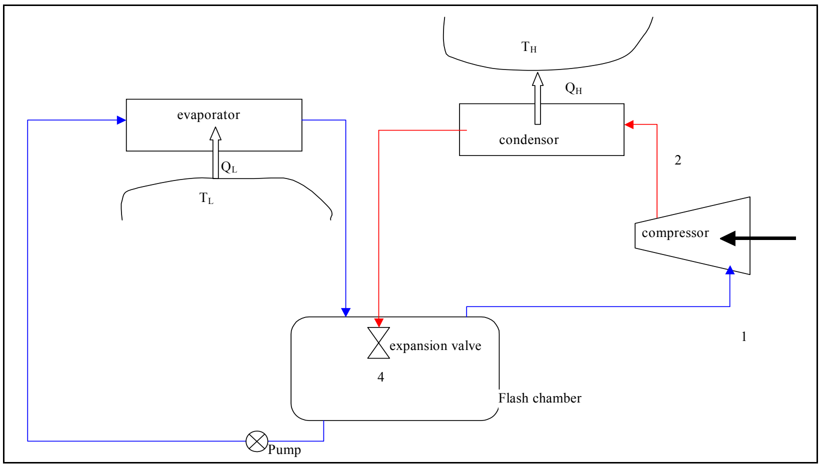

Figure 5: Schematic representation of ammonia refrigeration plant

The purpose of the flash chamber is to ensure that only saturated ammonia gas leaves the chamber before entering the compressor and that the saturated liquid left behind enters the evaporator, ensuring efficient pumping of the refrigerant.

In determining the coefficient of performance (COP_{RNH_3}) for the cycle shown in figures 5 and 6, the different mass flow rates through the evaporator and compressor must be known, as well as the quality and temperatures of the ammonia at the exit of the evaporator and compressor.

The flash chamber divides the cycle into two parts, the evaporation part and the condensing part. Each part has a different mass flow rate, which will affect the Carnot efficiency.

$$COP_{RNH_3} = \frac{Q_L}{W_{net,in}} = \frac{\dot{m}_{4f}\left(h_{evapout} - h_{evapin}\right)}{\dot{m}_{4g}\left(h_2 - h_1\right) + w_{pump}} \qquad (2.\ 6)$$

By using the coefficient of performance (COP_R) to determine the efficiency of the refrigeration cycle, the assumption has been made that the refrigeration cycle is an ideal vapour-compression cycle. In the latter refrigeration cycle, the flash chamber ensures that the compressor receives the refrigerant at saturated gas and that the evaporator receives the refrigerator at saturated liquid. This leads to different mass flow rates at the evaporator and condenser. The mass flow ratio has an effect on the COP_{RNH_3} for the flash chamber cycle from a Reverse Carnot heat engine viewpoint.

The temperature-entropy diagram for a simple vapour-compression refrigeration cycle with a flash chamber is shown below. The Reverse Carnot COP, which includes the flash chamber effect, is set out in the following paragraph. This efficiency is the ideal efficiency of the refrigeration cycle in place at Lambert's Bay and provides an indication of the maximum COP_R that can be achieved.

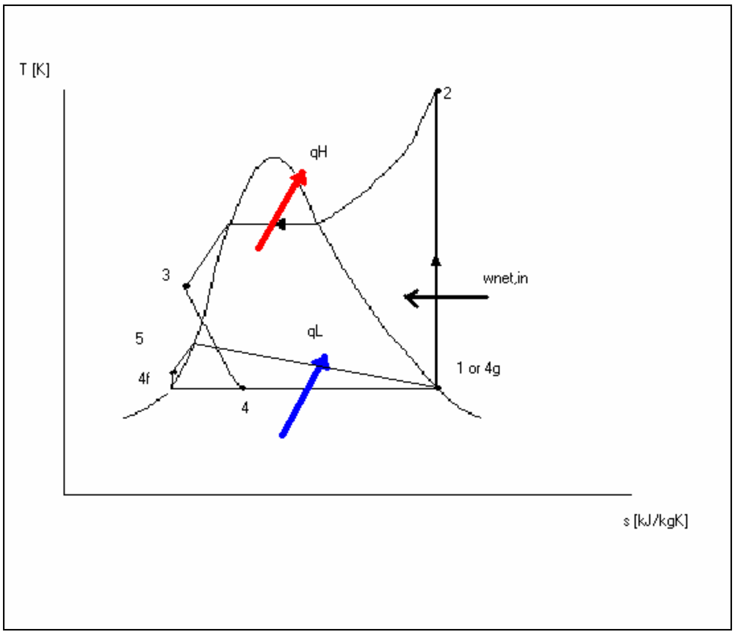

Figure 6: T-s diagram for vapour-compression cycle with flash chamber

The thermodynamic temperature scale used is the Kelvin scale. Its derivation is shown in Çengel, 1994. For a Reversed Carnot cycle, the following relation holds:

$$\frac{T_H}{T_L} = \frac{q_H}{q_L} = \frac{\dot{m}_f Q_H}{\dot{m}_g Q_L} \tag{2. 7}$$

The coefficient of performance for this cycle is:

$$COP_{RNH_3} = \frac{Q_L}{W_{net,in}} = \frac{1}{\frac{Q_H}{Q_L} - 1} \quad (2.\ 8)$$

From (2.7), the following can be derived:

$$\frac{Q_H}{Q_L} = \frac{\dot{m}_g T_H}{\dot{m}_f T_L} \quad (2.\ 9)$$

and therefore, the coefficient of performance for an ideal vapour-compression refrigeration cycle with a flash chamber is defined by:

$$COP_{RNH_3} = \frac{1}{\frac{\dot{m}_g}{\dot{m}_f}\frac{T_H}{T_L} - 1} \quad (2.\ 10)$$

From the derivation (equation 2.10) it is clear that, as the ratio of gas-to-liquid mass flow rate decreases, the coefficient of performance will increase.

To monitor the coefficient of performance, the mass flux at the outlets of the evaporator and the compressor needs to be measured and the source and sink temperatures at equilibrium must be measured for an accurate result. Furthermore, a central data processing point in the factory is needed where the data is sent to a computer for further calculations. To know the true coefficient of performance, the enthalpy of the refrigerant before and after the evaporator and compressor, as well as the different mass fluxes, are needed. The pump’s work must also be determined. This can be done electronically by measuring the current. However, all of these actions require a large amount of capital to install and are therefore not implemented in the industry.

Improving the Efficiency of the Refrigeration Cycle

By improving the energy efficiency of the different components in the refrigeration cycle, the efficiency of the entire refrigeration cycle will increase. The refrigeration plant consists of the following equipment: compressor, evaporator, condenser, valves and piping. The following section will discuss the different components in the refrigeration cycle and how to determine and improve their efficiency.

Compressor

A compressor is used as a means to move vapour or gas within a refrigeration system. The adiabatic efficiency of a compressor is defined as the ratio of the work input required to raise the pressure of a gas to a specified value in an isentropic manner to the actual work input. An enthalpy-entropy diagram of the process will show the different paths best.

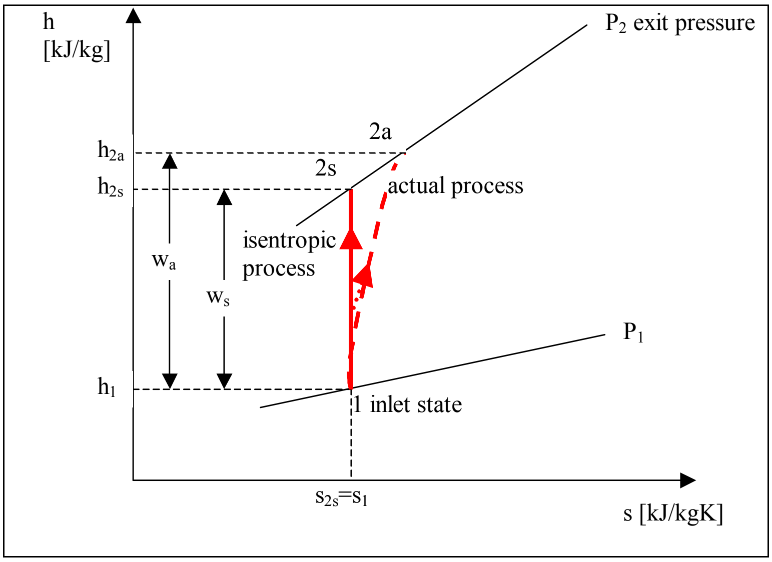

Figure 7: h-s diagram of an adiabatic compressor

The adiabatic efficiency of a compressor relates the isentropic work input to the actual work input:

$$\eta_C = \frac{isentropic\_compressor\_work}{actual\_compressor\_work} = \frac{w_s}{w_a} \qquad (2.\ 11)$$

By neglecting the changes in the kinetic and potential energy of the refrigerant being compressed, the efficiency can be obtained as a relation between an isentropic compression process enthalpy change and the actual compression process enthalpy change (figure 7):

$$\eta_C \cong \frac{h_{2s} - h_1}{h_{2a} - h_1} \tag{2. 12}$$

Determining the actual inlet and outlet enthalpies of the compressor, h_1 and h_{2a} respectively, is straightforward if the pressure and temperature is available. To find the ideal outlet enthalpy (isentropic path), h_{2s}, the thermodynamic property tables can be used.

Serious wear of cylinder walls and rings can result if an appreciable quantity of liquid returns with the suction gas. Large masses of liquid can damage the valves (for reciprocating compressors) or can cause more serious damage. Liquid returning steadily at rates too low to cause immediate evident damage will, however, reduce cylinder lubrication and cause extremely rapid abrasion of cylinder walls and rings. When the system has been installed carefully and properly with no foreign matter or liquid entering the compressor, it will operate satisfactorily for many years.

Evaporator

Losses at the evaporators can mostly be ascribed to fouling on the outside of the evaporator fins or inside the tubes. Whatever the losses, it can be determined if the overall design heat transfer coefficient, U_{design}, is known, and if the overall actual heat transfer coefficient, U_{actual}, which includes the effect of losses is known. By calculating the heat transfer to the refrigerant or air, $\dot{Q}$, the overall heat transfer coefficient can be calculated from the following equation:

$$\dot{Q} = UP\Delta x\left(T_H - T_L\right) \tag{2. 13}$$

Temperatures T_H and T_L are known and also:

$$\dot{Q} = \dot{m}\left(h_{out} - h_{in}\right) \tag{2. 14}$$

Therefore, in order to calculate the heat transfer rate at the evaporator, the mass flow rate must be measured and the enthalpy at the exit and entrance of the heat exchanger must be determined. From equation (2.14) the heat transfer rate, $\dot{Q}$, can be calculated and from equation (2.13) the overall heat transfer rate, *U*, can be calculated.

Proper evaporator piping and control are necessary to keep the cooled space at the desired temperature, as well as to adequately protect the compressor from surges of liquid ammonia from the evaporator.

Condenser

Induced draft cooling towers use the evaporation of water to condense the superheated refrigerant gas to a sub-cooled liquid. Fouling on the refrigeration pipes limits heat transfer to the water or air.

Fouling in or on refrigerant pipes leads to a decreased refrigeration capacity in the evaporator or condenser. Improving the heat transfer rate demands less work from the compressor for the same cooling capacity.

By increasing the heat load from the refrigerated area while the compressor work input remains constant, the coefficient of performance (COP) increases. This could be done by improving the performance of the cooling towers, or the heat transfer from the heat exchangers.

Oil in ammonia systems

Oil is miscible with liquid ammonia only in very small proportions. The evaporation of ammonia increases the oil ratio, causing the oil to separate. The increased density causes the oil to form a separate layer below the ammonia liquid, which can cover the heat transfer surface in the evaporator, thereby reducing performance. Therefore, oil must be removed periodically or continuously from the point where it collects. The service valves on the compressor open and close frequently and sometimes do not seat properly. Therefore, an additional stop valve can be used in the suction take-off to each compressor.

In addition, there should be oil separators in the discharge line of each compressor. A high-pressure float valve drains the oil back into the compressor crankcase or oil receiver. The oil separator should be as far as possible from the compressor to cool the discharge gas before it enters the oil separator. Liquid ammonia must not reach the crankcase.

Insulation of Refrigeration Piping

Insulation between refrigeration devices should be checked for any damage. By measuring the surface temperature of the insulation, the overall actual heat transfer coefficient for the insulation can be calculated and compared to new insulation.

Purging of Air in Ammonia

'Purging' is a term used to describe the process of removing unwanted air, vapours, dirt or moisture from the system. Air in a refrigeration system takes away some of its functioning capacity and failure to remove such air can be costly in terms of operating efficiency and damage to equipment. The heat transfer efficiency of any refrigeration system will greatly improve when undesirable, non-condensable gas is removed. This process of removing air is called purging.

No matter how hard one tries to avoid it, air will get into the system and accumulate on the inner surface of the heat exchanger, essentially creating an insulation barrier. Air could enter a refrigeration system through several mechanisms. When the suction pressure is lower than atmospheric pressure, air can enter through seals and valve packing. Air can also rush in when the system is opened for repair, coil cleaning, when equipment is added, when the refrigerant truck is charging the system or when oil is added. This accumulated air insulates the heat transfer surfaces and, in effect, reduces the size of the condenser. To offset this size reduction, the system must work harder by increasing the pressure and temperature of the refrigerant. Therefore, prompt removal of air is essential.

Before air can be removed, one must determine where it will collect. With multiple condensers and receivers it could be difficult to determine the exact location of the air. Condenser piping design, component arrangement and operation affect the location of air. It is important to frequently purge each suspected air purge point (one at a time) to ensure all the air is removed from every location.

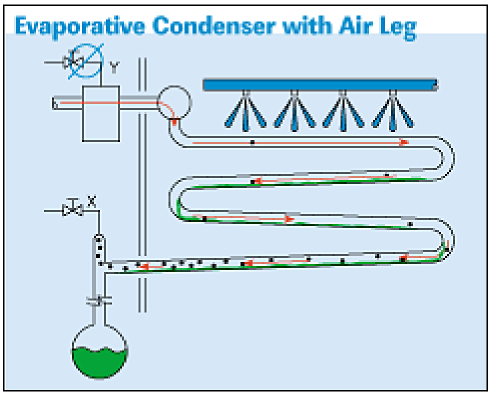

Figure 8: Evaporative condenser
(Rockwell & Quake, 2001)

With an evaporative condenser (figure 8), the velocity of the entering gas prevents any significant air accumulation upstream from point X. High velocity past point X is impossible, because the receiver pressure is virtually the same as the pressure at point X. To ensure that air accumulates and move into the purger, an air leg is recommended. The purge connection on a receiver should be at the point furthest from the liquid inlet.

The easiest way to estimate the amount of air in the refrigeration system is by checking the condenser pressure and the temperature of the refrigerant leaving the condenser. By comparing the findings with data provided in a temperature-pressure chart, such as the one published in the ASHRAE, 1997 Fundamentals Handbook, the extra cost caused by air accumulation from compressor operation can be estimated. For each 27.58 kPa of excess head pressure caused by air, the power cost to operate the compressor increases by 2% and the compressor's capacity is reduced by 1% (Rockwell & Quake, 2001).

Removal of Contaminants in the Refrigerant

Ammonia is a powerful solvent which removes dirt, scale, sand or moisture remaining in the pipes, valves and fittings during installation. These substances are swept along with the suction gas to the compressor, becoming a danger to the bearings, pistons, cylinder walls, valves and lubrication oil.

Operating temperatures in actual cycles are established to suit the temperatures required at the cold medium and the temperature acceptable for the heat sink. The practical temperature gradient required to transfer heat from one fluid to another through a heat exchanger, falls in the range between 5 ^{o}C and 8 ^{o}C. This leads to an increase in the work required to produce the desired refrigeration effect, because the temperature difference has increased.

In the refrigerant cycle, the refrigerant gas is superheated at both the evaporator and the compressor. During the evaporation process, the refrigerant is completely vaporised partway through the evaporator. As the cool refrigerant vapour continues through the evaporator, additional heat, which superheats the vapour, is absorbed. Pressure losses caused by friction furthermore increase the amount of superheat. When superheating of the refrigerant occurs at the evaporator, the enthalpy of the refrigerant is increased, additional heat is extracted and the refrigeration effect of the evaporator is increased. No useful cooling takes place when superheating occurs in the compressor suction piping. The increased refrigeration effect in the evaporator is usually offset by a decrease in refrigeration effect in the compressor. This occurs because the volumetric flow rate of a compressor is constant, and therefore the mass flow rate and refrigeration effect are reduced by a decrease in refrigerant density caused by superheating.

Refrigerant superheating also occurs at the compressor. Under ideal conditions, the refrigerant enters the compressor as a saturated vapour. Increasing the pressure will increase the temperature and cause superheating. Superheating caused by the compression process will not improve cycle efficiency, but will result in bigger condensing equipment and large compressor discharge piping.

De-superheating is the process of removing excess heat from the superheated refrigerant vapour and, when accomplished through means external to the cycle, it could be beneficial to system performance. Also, de-superheating the high-pressure refrigerant (hot gas) that leaves the compressor, will reduce the required condenser capacity and provide a high-grade heat source for other process uses. A typical application would be the preheating of boiler makeup water or process water.

Liquid sub-cooling occurs when a liquid refrigerant is cooled at a constant pressure below the liquid's condensation temperature. When sub-cooling occurs through a heat transfer method external to the refrigeration cycle, the refrigeration effect of the system is increased, because the enthalpy of the sub-cooled liquid is less than the enthalpy of the saturated liquid. Sub-cooling of the liquid upstream of the throttling device also reduces flashing in the liquid piping. The work input is reduced and the refrigeration effect increased.

Hot gas bypass is a method of placing an artificial heat load on the refrigeration system in order to produce stable suction pressures and temperatures when the refrigeration load is very low. Bypassing hot gas from the compressor discharge to the evaporator inlet or compressor suction produces the heat load. Although hot gas bypass permits stable compressor operation at low load, it wastes energy.

When a refrigerant system operates with the evaporator temperature close to 0 ^{o}C or less, freezing of the evaporator coil is inevitable. Defrosting accessories are available from the manufacturers for systems operating under these conditions.

Cold Store Doors

A lot of energy and time is spent to produce cold air from hot humid air to temperatures below 0 ^{o}C. The cold storage door plays an important role in today's cold storage warehouse and the reasons are obvious when you look at some of the conditions currently faced by warehouse operators and owners.

Stewart (2001) wrote an article on the effect of insulated door openings in cold storage warehouses, in which some points that affect modern cold storage doorways were put forward. Below follows a summary of these points.

Energy costs continue to consume a significant portion of a warehouse's operating budget. Approximately 10% of a warehouse's operating budget is tagged to the electricity account. Changing product mix and customer quality demands prompted many facilities to take temperatures below the former 0 ^{o}C benchmark. Furthermore, these temperatures had to be held consistent through the distribution chain.

More goods are flowing through warehouses. Studies showed that approximately 46% of the warehouses worldwide are seeing increased inventories, with approximately 42% reporting higher inventory turnover during 1998. Facility material handling systems will have to keep abreast with the pace. A high-density freezer store/deepfreeze cold storage warehouse is the most expensive type of storage to operate. The solution is high-density storage in the form of mobile racking systems to provide maximum space utilisation, access to every location ('First In, First Out'), least product damage, greater flexibility, a simple operating system, lowest initial capital cost per useable location and greatest income potential for any given size of building.

Faster material-handling equipment, such as powered pallet truck trailer-to-transport pallets from road vehicle over dock leveller into the drop temperature area, cause bottlenecks through the insulated doorway into the deepfreeze area.

It is a fact that, for productivity reasons, cold storage doors are left open. With traffic entering and leaving the cold storage rooms, opportunities for air transfer are greater and result in a vast amount of wasted energy. In order to take corrective action, the open doors will have to be monitored more fully and understood better by those in charge of the warehouse operation.

The design and location of cold storage door systems should encourage the fleet of forklifts to move as efficiently as possible (on average, forklift drivers can spend up to 50 seconds going through a door).

The efficient operation of any refrigeration system is faced with two problems: infiltration and frost. The following section will discuss how infiltration and frost can result in excess, but avoidable, costs and how cold storage doors can help one control the escalation of these costs.

A large part of the refrigeration capacity at cold stores is wasted on the freezing of water due to an increase in the humidity contents of air when product handling requires opened doors. The forming of frost on the evaporator surface insulate the surface and decrease the overall heat transfer coefficient of the evaporator. This leads to inefficient heat removal in cold stores.

The removal of moisture in cold stores is therefore an option for more energy-efficient cold storage operation. The drying of air (removal of moisture) can be done in many ways. The purpose is to remove moisture from air in the regions in front of the door that are most likely to enter the cold store as the door opens. A buffer region can be formed in front of the door by means of rubber curtains. Air is taken from just before the buffer region and the drier air is then forced into the buffer region, causing a positive pressure and ensuring that only dry air enters the cold stores. The drying of air with dehumidifiers can occur in two ways: absorption dehumidifying and desiccant dehumidifying.

With absorption dehumidifiers, a hygroscopic salt solution (see figure 9) is pumped around and sprayed into the dehumidifier (A). Humid air (from outside or recycled air) passes into the dehumidifier. This air comes into close contact with the hygroscopic salt solution spray, which absorbs the moisture present in the air. Dry air leaves the top of the unit and the salt solution, including the absorbed moisture, collects in the lower part of the unit. By subsequently cooling the salt solution, the air is cooled and dried simultaneously. Drip catchers at the air outlet of the dehumidifier ensure that the air stream does not contain any particles salt solution.

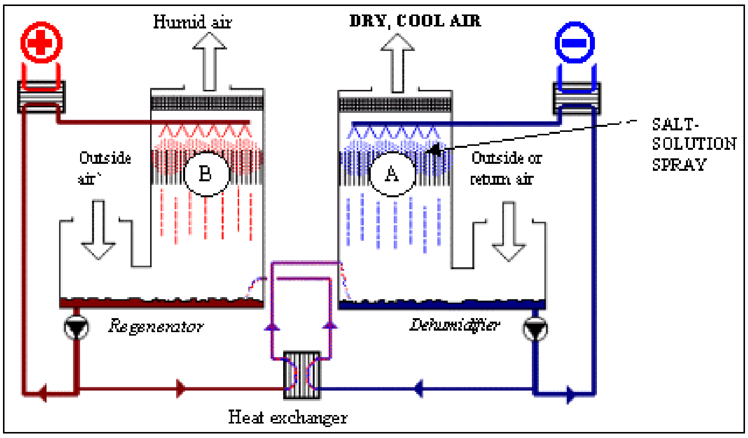

Figure 9: Principle of operation of the absorption dehumidifier

(http://www.kathabar.imtech.nl/)

To ensure a stable concentration of salt in the dehumidifier, the absorbed moisture has to be evaporated. Therefore, a part of the (diluted) salt solution is pumped to the regenerator (B). Here, the hygroscopic salt solution is pumped and sprayed around again. At the regeneration side, water is evaporated by heating the salt solution. A minor secondary air stream passing through the regenerator absorbs this moisture and takes it outside. The concentrated salt solution returns to the dehumidifier. The process flow diagram indicates that the cold (diluted) salt solution from the dehumidifier meets the ‘warm’ salt solution in the regenerator. A heat exchanger, placed between these flows, will preheat the ‘cold’ solution before it enters the regenerator. The warm solution will radiate heat, hence the cooling down before it is used in the dehumidifier again.

Desiccant dehumidifiers utilise a ‘sorption’ material to attract and hold moisture from air. Once the sorption material (called a desiccant) is ‘saturated’ with moisture, it can be reactivated or regenerated. Reactivation is usually accomplished by thermal means and restores the desiccant's dehumidification capacity. The mass exchange of the moisture from and to an air stream occurs in the vapourisation phase.

Desiccant dehumidifiers are required for use below the frost point where mechanical refrigeration type dehumidifiers experience freezing on the coil surface or when dehumidification is required, but cooling is not, such as for dry goods storage or preservation requirements. Desiccant dehumidification is also utilised to provide for humidity control independent of temperature control in occupied spaces.

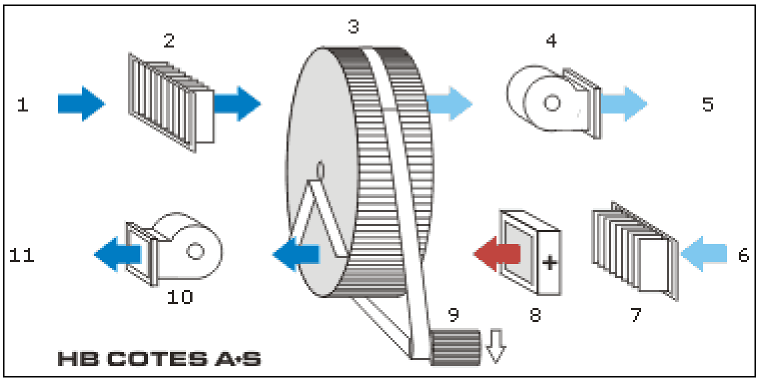

Figure 10: Dessicant dehumidifier operation

(http://www.hbcotes.com/english/home.html)

Doors have to be opened and products have to be moved in and out of the cold storage space to justify the cost of the space. In a busy facility, this occurs hundreds of times per day. Every time the doors are opened, two air streams are set in motion at the doorway, namely cold, dry air that escapes from the room along the floor and warm, moist air that enters through the upper section of the door.

The warm, moist air stream almost immediately migrates to the evaporator coils to form frost. This air infiltration can account for more than half the total refrigeration load of a cold storage facility. The role of the evaporator coil in the refrigeration process is to draw heat and moisture from the air, supplying cold and dry air resulting in and maintaining low temperatures within the cold storage room. Additional units may be out in the dock-drop temperature area to cool and dehumidify air before it enters the cold storage room and to keep products cold during loading and unloading, or otherwise dehumidify and heated through mechanical dehumidifiers.

If the product entering the room has not yet been pre-cooled, it can allow considerable quantities of heat and moisture to enter with the warm air through the open door. The (condensed) moisture drawn from the air accumulates on the evaporator coil and turns into frost. As more moisture enters the cold storage room with each opening of the door, so the frost layer grows and acts as an insulator. System efficiency therefore degrades, ultimately causing the unit to become inoperative. When the refrigeration system or the person responsible for operating the system detects the frost build-up, the refrigeration process is suspended and heat must be applied to the coil to melt the frost (called the defrost cycle).

Fortunately, the full effect of this air exchange does not occur immediately as the door is opened. The rush of the door opening, traffic going through the door and differential pressures will initially disrupt the course of the airflow, causing the warm and cold air streams to mix at the doorway. There are approximately 15 to 60 seconds to close the door before the air streams reach a steady state (still air). Preventing or minimising the ability of the air streams to reach a steady state means reducing the open door time. Based on a time study, opening and closing a manual door for an efficiently run facility is close to 120 seconds. Modern high-speed, horizontal-opening doors open at a speed of 1.1 meter per second. Therefore, to open and close a door would be approximately 6 seconds.

Telltale signs of losses in temperature and financial terms are the noticeable wet floor, fog (mist) outside the deep freeze entrance and the slippery ice on the inside caused by heavy, cold air escaping from the deep freeze. The longer the door is open and more cold air flows out at the bottom and creates a vacuum at the top to be filled by warm air that flows in at the top, the more energy is lost. It is therefore essential to stem the cycle through a faster, automatic high-speed door.

Air curtains (air knives) are also an alternative option to a door (see figure 11). Air curtains minimise the amount of water vapour travelling from the high vapour pressure region (outside the cold store) to the low vapour pressure region (inside the cold store). This phenomenon is known as water vapour infiltration and increases the latent heat load of the cold stores. Another unwanted occurrence is the formation of frost on the evaporator coils, which obstructs the airflow through the coil and needs to be defrosted by bypassing superheated gas through the evaporator coil. The air curtain limits water vapour infiltration by approximately 70% – 90% of the amount that will infiltrate without any obstruction. Applying the use of air curtains where high vapour infiltration occurs will lead to significant energy savings.

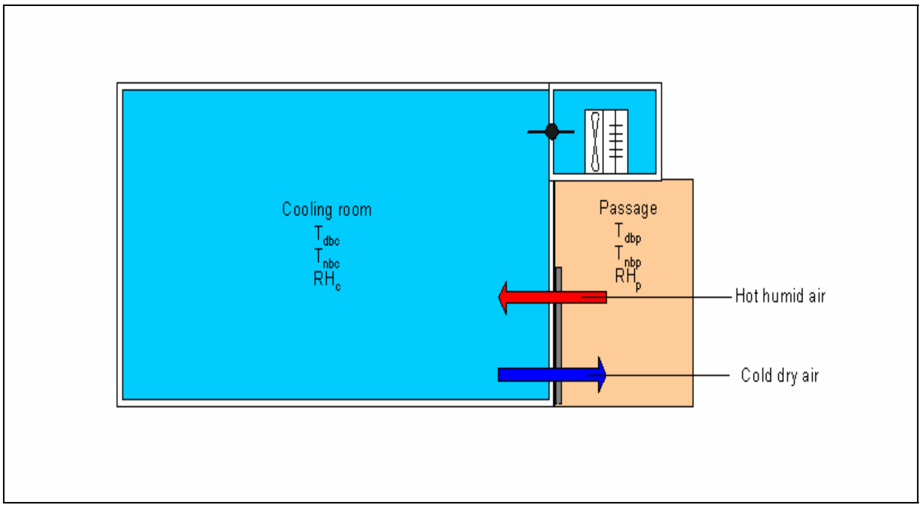

Figure 11: Infiltration

In addition to the energy cost connected with the build-up of frost on the evaporator coils, there is the cost of defrosting. Theoretically, defrosting would require about 50% – 100% of the energy needed to make the frost in the first place. Conducting an energy audit for cooling rooms are unique to each situation, but there are certain similarities. Most cold stores have the problem of moist air entering the room and thus increasing the latent heat load inside the room. Factors that need to be considered are the volume of the room, the surface area of the walls and the surface finish, the heat load that needs to be removed per unit time, the surfaces covered by frost and an approximate volume of the ice, as well as the average dry and wet-bulb temperatures of the room. An energy balance will identify the amount of heat load contributed by the moist air (Stewart, 2001).

2.2.6 Measuring Instrumentation

In determining the energy efficiencies of different equipment or systems, it is necessary to determine the qualitative and the change in quantitative properties of the working fluid or fluid under consideration. The quantitative properties are mostly mass flow and can be obtained by measuring the volume flow rate (density must be known) or mass flow rate of the fluid. The qualitative properties can be obtained from pressure and temperature measurements. A criterion that the measuring instruments must satisfy is that the measurements must be electronically stored to enable long-term (seasonal) monitoring.

Flow meters are used to monitor and measure volume and mass flow rates and are divided into five basic categories, based on the method of measurement (Bagajewicz, 2001):

i. Differential pressure meters
ii. Velocity meters
iii. Positive displacement meters
iv. Mass meters
v. Variable area meters

Several factors play a role in the selection of the best flow meter for a particular application, for instance fluid properties, desired performance, method and/or ease of installation, safety, environmental impact and cost (often the overriding factor).

Differential pressure meters consist of three general types, namely orifice meters, venturi meters and flow nozzles. Orifice meters are the most widely used differential pressure meters. To maintain accuracy, they require 5 – 40 straight pipe diameters upstream of the meter. These flow meters have many advantages, which include durability, low cost for large pipe diameters and the ability to be used at a variety of temperatures. The disadvantages are their high sensitivity to density and viscosity changes, and a tendency to erode. Venturi meters are used in high flow rate applications and are more expensive than orifice meters. However, in high flow rate applications they may cause significant pressure losses and diminished accuracy. Flow nozzles are used when capacity is an important issue and they work well in high-velocity, high-temperature and high-turbulence situations. Flow nozzles are less expensive than venturis, but result in higher permanent pressure losses and are less accurate.

Velocity meters measure fluid velocity and use the relationship $Q = v \cdot A$ to calculate volumetric flow rates, where Q is the volumetric flow rate, v the fluid velocity and A the flow meter cross-sectional area. There are five basic types of velocity meters, namely electromagnetic, vortex, turbine, ultrasonic and pitot meters. Fluids used in electromagnetic meters must be electrically conductive and non-magnetic. Because there are no mechanical parts that impede fluid flow, electromagnetic meters are excellent for streams containing particles or corrosive chemicals. Other advantages include their ability to provide accurate voltage signals in laminar and turbulent flow, as well as their low sensitivity to changes in density, pressure and viscosity. A disadvantage of these meters is the potential of the electrodes to become coated, thereby reducing errors in the reading. However, this can be eliminated by keeping flow between 2.44 m/s and 6.05 m/s.

Mass flow meters are classified in four types, namely Coriolis, thermal, heated element and temperature rise mass flow meters. Coriolis meters are the most accurate instruments and, until 1999, were mainly used in critical control loops and for the management of high-valued fluids. Advantages include their high accuracy, large flow range and tight control ability. In addition, they do not obstruct the fluid, do not need recalibration and can be used with different types of fluids. Thermal mass flow meters are effective with clean, low-density gases. They are used to measure gas flow in ducts, pilot plants, purge streams and leak testing, and are most commonly used in "dopant" gas flows for semiconductor production. One of the advantages of a thermal mass flow meter is high accuracy. A disadvantage is the coating of probes by dirty fluids and the need for uniform fluid flow.

Temperature measuring devices are divided into three groups:

i. Thermal expansion thermometers
ii. Electrical devices
iii. Radiation-based devices

Thermal expansion thermometers are based on the thermal expansion of either liquids or solids. Many thermal expansion thermometers are relatively inexpensive, but they are not very compatible with the modern computer-based data acquisition systems. Advantages are their simplicity and their low cost. Disadvantages include that they need to be repaired at the factory and calibration changes with handling. Electrical devices include thermocouples, thermo resistance meters and thermistors. Of these, thermocouples are the less expensive and more popular electrical device. A disadvantage, however, is the cold junction that may affect calibration. Thermo resistance meters are accurate, but more expensive than thermocouples, while thermistors can work over a very narrow span with no cold junction. A disadvantage is their non-linear response.

Process pressure measuring devices are divided into three groups:

i. Liquid column devices
ii. Elastic element devices

iii. Electrical sensing devices

With liquid column devices, pressure is measured by determining the height of a liquid column. If the density of the liquid is known, this height is a measurement of the pressure. Most types of liquid column devices are commonly called manometers. There are three types of elastic element pressure measuring devices, namely Bourdon-tube, bellows and diaphragm elements. These devices rely on the deformation of an elastic material that is proportional to the applied pressure. Electrical sensing devices operate on the property that electrical resistance of conducting solids changes with diameter and length.

Monitoring and analysing energy consumption of different equipment and systems at once over a certain period require effective and fast conveying of measurement signals to processor. This requires the measurement signals to be conveyed electronically. Devices performing this operation are called transducers. Ideal transducers are linear, but in practice they are affected by zero offsets, hysteresis, et cetera. By using transducers the measurement can be transformed into an electronic signal and sent to a processor where the data is processed.

3 LAMBERT'S BAY ENERGY ANALYSIS

3.1 Energy Analysis Strategy

The energy analysis that was carried out consisted of a walkthrough audit, an energy consumption analysis, an energy efficiency analysis, an energy efficiency upgrading analysis and a payback period calculation. An energy efficiency study was implemented at Lambert's Bay Foods with minor adjustments to adapt to Lambert's Bay's circumstances and financial limitations. Therefore, this study focused on identifying losses and suggesting solutions to minimise the losses in order to improve the thermal efficiency of the processes or equipment.

3.2 Plant Layout

The plant comprises a French fries production line, three cold stores, a blast freezer, a compressor room, a loading bay and a boiler room. All the abovementioned is shown on the schematic drawing in figure 12, except the boiler room, which is situated approximately 30 m from the production line in another building adjacent to the French fries production line.

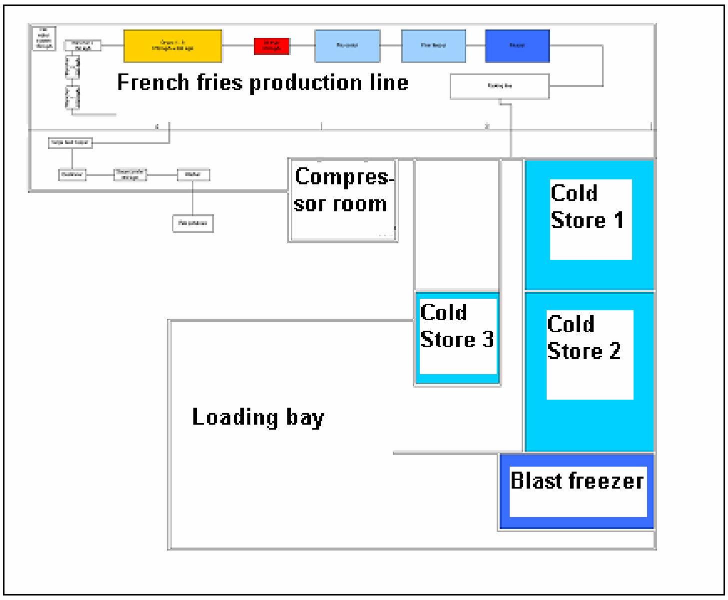

Figure 12: Schematic representation of plant layout

3.3 Walkthrough Audit

A walkthrough audit checklist was generated from a checklist suggested by Kreith and West (1997) – see appendix A.4. In June 2001, a walkthrough audit of the premises was conducted to identify energy wastages and possibilities of energy efficiency improvements (see appendix A.4).

Visible signs of energy wastages in the form of ice were found on the uninsulated fittings of the refrigeration pipe network, as well as on the uninsulated valves and similar items of the refrigeration system inside the refrigeration room. Calculations were conducted to estimate energy losses on the refrigeration pipes.

Steam leaks were also detected at various locations in the plant. Visible steam losses are at the steam peeler, the valve near the hot water reservoir and at the blanchers. In the case of the steam peeler, the same situation occurred at a French fries plant in the Netherlands (Dutch National Team, 2000). There, the discharge steam was recovered by means of a heat exchanger.

Huge potential for energy savings exists by modifying the production line. By increasing the production line efficiency, less energy is wasted by adding value to defective products that are thrown away. A large amount of refrigeration energy is wasted at the flow freezer, where the frozen French fries are manually sorted and exposed to a warm environment (a human being requires a comfortable working environment). The temperature increase caused by the exposed fries needs to be corrected by putting the packed fries into a blast freezer, rapidly cooling the fries to –37 ^{o}C. An optical sorter can be used to shorten the length of exposure of fries to the warm environment (temperatures close to 0 ^{o}C). Automating the handling of material from the packing line to the cold stores means that the environment in which the fries are packed and transported can operate at a much lower temperature. This, however, requires a thorough feasibility study to motivate the capital needed for such procedures.

3.4 Energy Consumption

In order to determine the feasibility of an energy analysis, knowledge of the main energy-consuming equipment or processes is essential. Energy accounts for six months (December 2001 to May 2002) were used to compare the main energy-consuming equipment (see appendix A.1). By using an average price for electricity, assuming the plant operates 24 hours a day, coal and electricity consumption can be compared on a cost basis. From the obtained accounts, coal is accountable for 79% and electricity for 21% of the total energy costs. The energy cost for coal per joule is R0.0157 and for electricity R0.0446. The different steam users are indicated in figure 13, with its fraction of total steam consumption (data obtained from Uticon-Dynatherm Industrie bv, 1994).

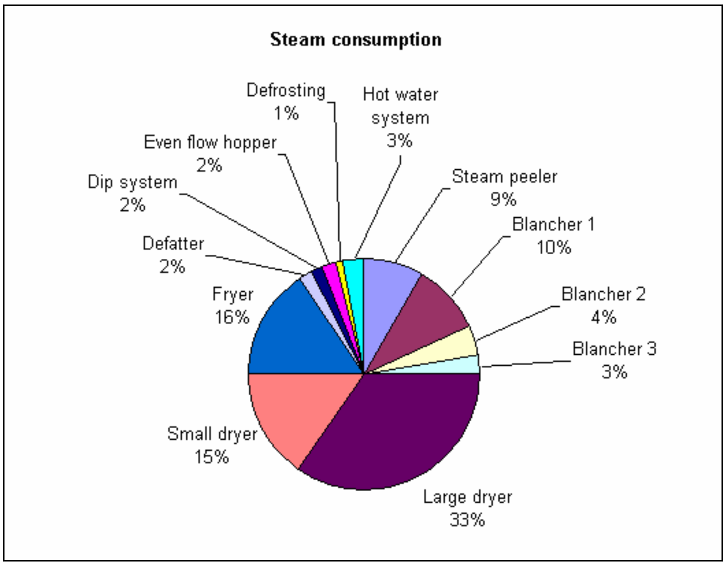

Figure 13: Pie chart indicating the main steam users

The largest steam user is the large dryer, which consumes 33% of the total steam consumption. The fryer consumes 16% of the total steam production to fry the cooked fries. Other large steam consumers are blanchers 1, 2 and 3 (10%, 4% and 3% of total steam production respectively), the small dryer (15%) and the steam peeler (9%).

The electricity usage can also be compared in terms of equipment and is indicated in the pie chart below.

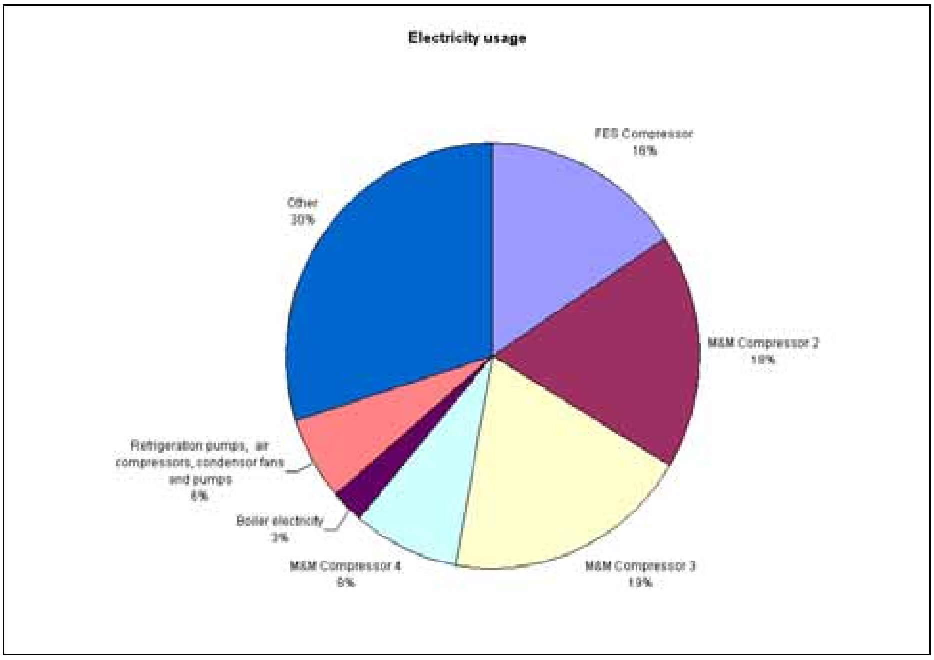

Figure 14: Electricity consumption of various users

From figure 14, it can be seen that refrigeration contributes 67% (FES compressor, M&M compressors 2, 3 and 4, refrigeration pumps, air compressors, condensor fans and pumps) of the total electricity consumption, which makes it the largest user of electricity. The consumers of refrigeration energy can furthermore be divided into different equipment.

The data from the above figures was used to draw up a Sankey diagram (figure 15) to indicate the energy flow graphically.

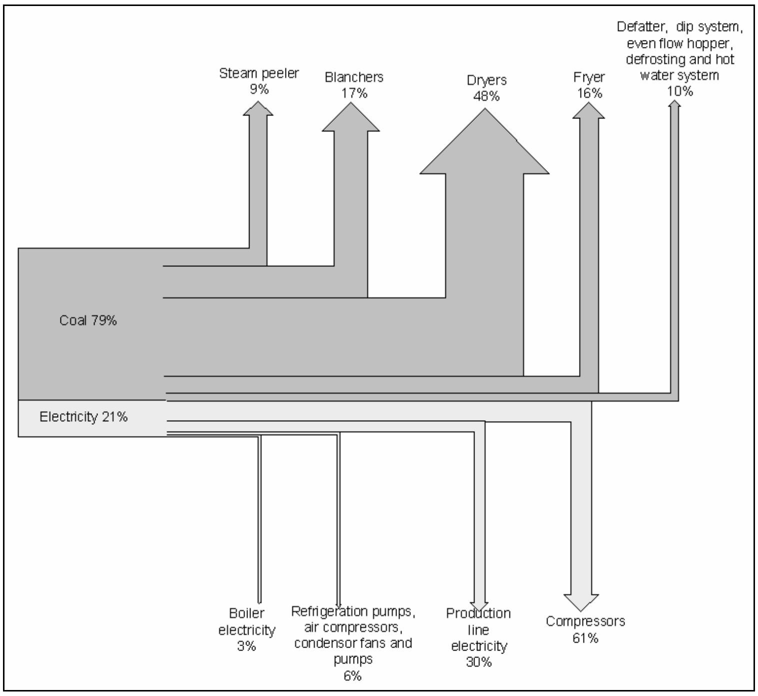

Figure 15: Sankey diagram of total energy flow in plant

From the Sankey diagram, the dryers and blanchers are the largest users of energy and then the compressors and fryer follow. The Sankey diagram graphically helps to establish focus areas in an energy efficiency analysis.

3.5 Economic Decision Analysis Approach

Where the payback period is so long that inflation needs to be considered, an actual Rand approach will be used. Assumptions made to simplify the economic analysis were a constant interest rate and, that for short payback periods (less than five years) the inflation effect is insignificant, therefore, the payback amount will be the energy savings that were achieved. Other costs, such as maintenance, will be included in the overall cost of the equipment where possible. Fabrycky, Thuesen and Verma (1998) offer the following formula which relates the present value, single payment per interest period and the number of interest periods:

$$P = \frac{A \cdot \left((1+i)^n - 1 \right)}{i \cdot (1+i)^n} \qquad (3.\ 1)$$

After manipulation, the number of interest periods (which, in our case, will be monthly) can be calculated:

$$n = \frac{\ln\left(\dfrac{1}{1 - \dfrac{P}{A} \cdot i} \right)}{\ln(1+i)} \qquad (3.\ 2)$$

In most cost analyses in this document *n* is the number of months, *P* the initial capital outlay and *A* the monthly energy savings.

3.6 Steam

3.6.1 Dryers

The dryers, which consume 48% of the total steam production, were analysed to measure their performance. The function of the dryers is to control the moisture content of the fries. This is achieved by removing moisture from the fries by blowing warm, dry air through the fries as they are transported on a chain conveyor belt. Air is sucked in (by centrifugal fans) from outside the plant and heated with steam-to-air heat exchangers to decrease the relative humidity of the air. The air moves from right to left through the dryers as shown in figure 16 and the fries move on the chain belt in the opposite direction. The low relative humidity of the air improves the mass transfer potential of moisture from the fries into the air.

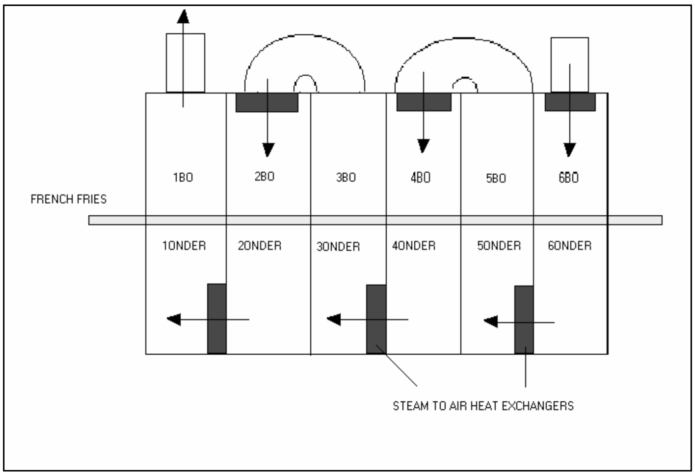

Figure 16: Schematic drawing of dryer illustrating airflow

Measurements of the surface temperature of the fries and the wet and dry-bulb temperatures of the air in each section of the dryer were taken (appendix J.1) in order to calculate the relative humidity of the air inside the dryers. The relative humidity difference between the air and the moisture just above the surface of the fries is the driving force for evaporation. It was found that one of the steam-to-air heat exchangers in the dryer was performing inadequately, which limited the current capacity of the dryer and prohibited an increase in production rate.

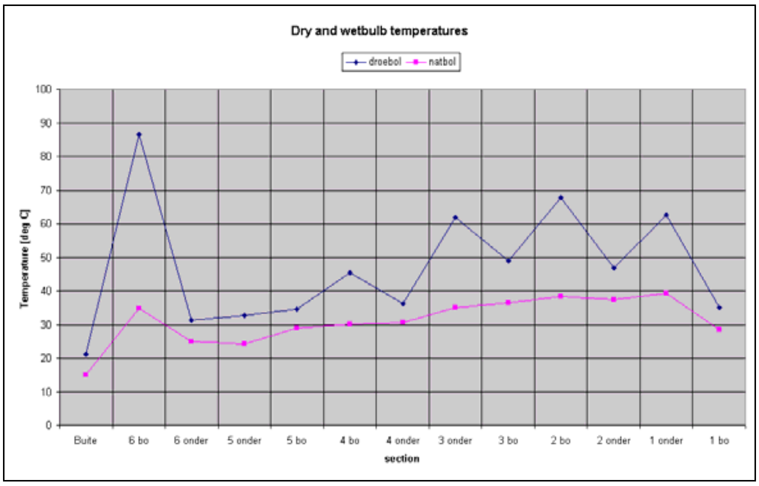

Figure 17: Dry and wet-bulb temperatures of each section

As indicated in figure 17, the outside air is at a relative low dry and wet-bulb temperature. As soon as the air moves through the first heat exchanger, the dry-bulb temperature is supposed to increase and the wet-bulb temperature is supposed to stay constant. However, it was found that the wet-bulb temperature increased with the dry-bulb temperature and that there was no airflow through the first heat exchanger. Further investigation showed that air from the plant is sucked in from the right side of the dryer between the belt and the opening. This action prohibits the first heat exchanger from being fully utilised, thereby limiting production increases and leading to inefficient use of centrifugal fans.

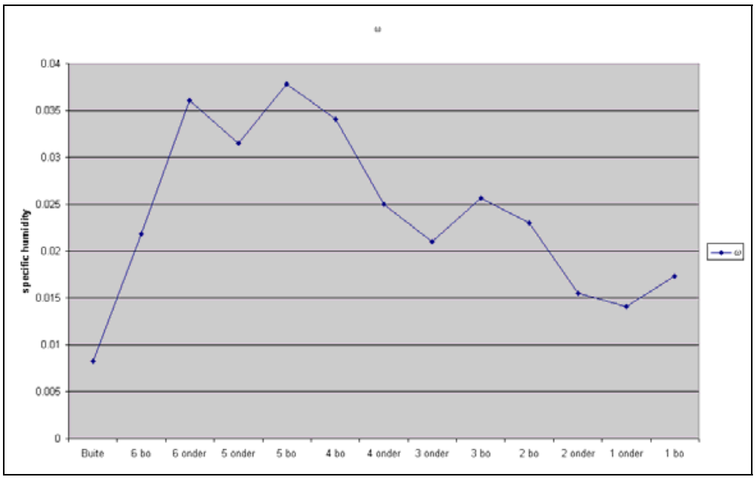

Figure 18: Specific humidity of air through each section

Figure 18 indicates the specific humidity of air at different sections in the dryers. This is an direct indication of the amount of humidity in the air which must increase right through the dryers until the air exits at 1 BO. This figure suggests that dryer air from outside mixes with the main air stream, thus showing a decrease in specific humidity.

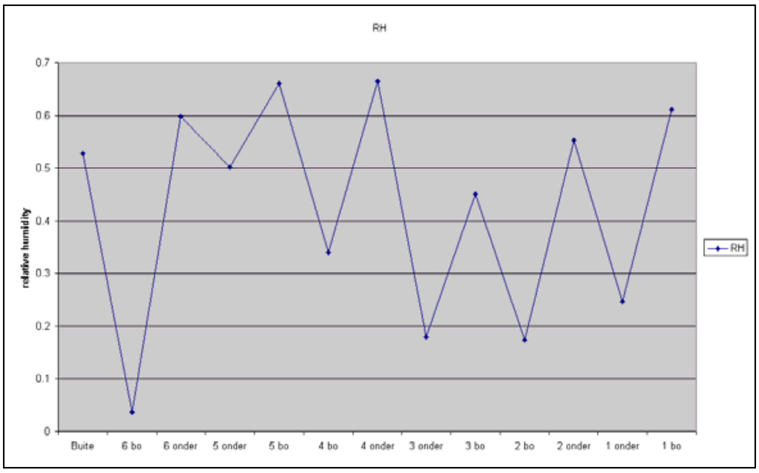

Figure 19: Relative humidity of air through each section

Figure 19 shows the relative humidity for each section and looks very favourable. This is exactly what is needed for moisture removal. If only the airflow through the heat exchangers was favourable and the heat exchangers were performing to specifications, the dryer's capacity would increase substantially.

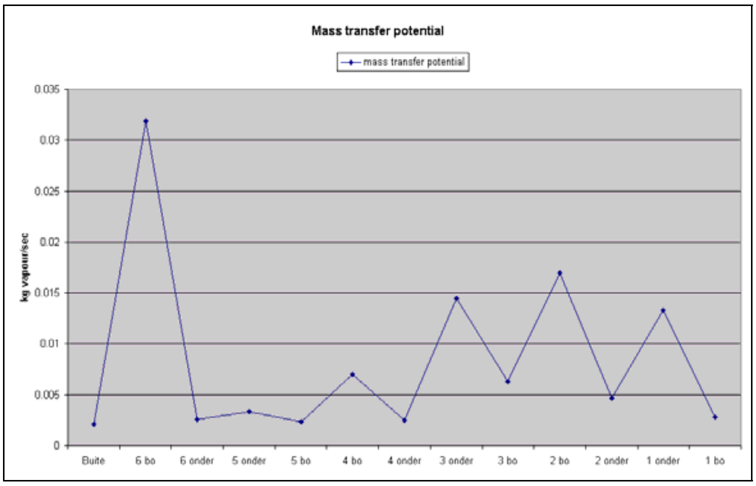

Figure 20: Mass transfer of water from fries to air

The arrows in the above figure indicate the relative mass transfer potential at each point before the fries. From this figure, it is obvious that at 6 BO the air offers the largest potential to remove moisture from the fries. However, inspection showed that very little of the air at 6 BO moves through the fries due to fouling on the heat exchanger fins. Closer investigation identified large volumes of air entering from the right-hand side of the dryer through a narrow opening where the fries chain belt exits. This occurs, because the pressure difference required for the air to move from 6 BO to 6 ONDER is much higher than for the air to travel from the outside of the plant to 6 ONDER. Another possibility is that the channel through which air moves from the outside to 6 BO is blocked. Either way, it is advisable that the channel, as well as the heat exchanger, is inspected and cleaned on a regular basis and that the pressure difference needed for airflow through the conveyor opening is increased. The latter can be achieved by adding an additional channel and obstruction section over the conveyor. Figures 18 to 20 were obtained through spreadsheet calculations in appendix J.2.

Calculations to Estimate Specific Humidity and Mass Transfer

Calculations to determine the specific humidity from the dry and wet-bulb temperatures in figure 17 were carried out in an Excel spreadsheet (appendix J.2). For determining the specific humidity, one looks at an adiabatic saturation process where the incoming air is at given dry and wet-bulb temperatures and the outlet air is saturated at a dry-bulb temperature equal to the inlet wet-bulb temperature.

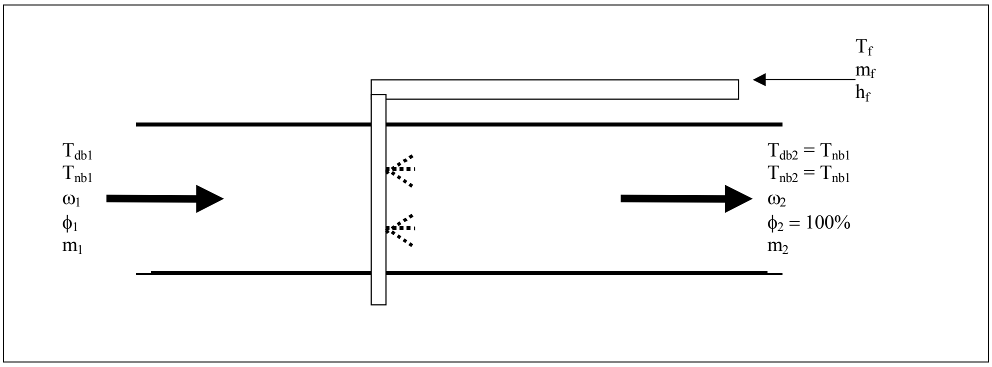

Figure 21: Adiabatic saturation of air

In the schematic illustration in figure 21 air at known dry and wet-bulb temperatures is saturated by adding water spray into the air. Air enters at T_{db1}, T_{nb1} with a mass flow rate of m_1, water is sprayed at a temperature of T_{nb1} and a flow rate of m_f and it exits at $T_{db2} = T_{nb1}$, $T_{nb2} = T_{nb1}$ and a total mass flow rate of m_2. The mass flow rate of dry air remains constant at m_a. From the conservation of mass for water vapour we have:

$$\omega_1 \cdot m_a + m_f = \omega_2 \cdot m_a \qquad (3.3)$$

In equation 3.3, m_a is known, but the rest are all unknown. The saturated vapour pressure at the exit can be calculated for the wet-bulb temperature, T_{nb1}, from Kröger (1998):

$$P_{g2} = 10^{z_2} \tag{3. 4}$$

where z_2 is a function of the wet-bulb temperature:

$$\begin{aligned} z_2 &= 10.79586 \cdot (1 - \frac{273.16K}{T_{nb1}}) + 5.02808 \cdot \log(\frac{273.16K}{T_{nb1}}) \\ &+ 1.50474 \cdot 10^{-4} \cdot \left[1 - 10^{-8.29692 \cdot (\frac{T_{nb1}}{273.16K} - 1)} \right] \\ &+ 4.2873 \cdot 10^{-4} \cdot \left[10^{4.76955 \cdot (1 - \frac{273.16K}{T_{nb1}})} - 1 \right] + 2.786118 \end{aligned} \tag{3. 5}$$

With P_{g2} known and the relative humidity $\phi_2 = 1$, the specific humidity at the exit can be obtained from:

$$\omega_2 = \frac{0.622 \cdot \phi_2 \cdot P_{g2}}{P - P_{g2}} \tag{3. 6}$$

where P is atmospheric pressure. From the conservation of energy:

$$m_a \cdot (h_{a1} + \omega_1 \cdot h_{v1}) + m_f \cdot h_f = m_a \left(h_{a2} + \omega_2 \cdot h_{v2} \right) \tag{3. 7}$$

Through manipulation the specific humidity at the inlet is:

$$\omega_1 = \frac{h_{a2} - h_{a1} + \omega_2 \cdot \left(h_{v2} - h_f \right)}{h_{v1} - h_f} \tag{3. 8}$$

The specific enthalpy of dry air (approximated air as an ideal gas), h_a, can be obtained as follows:

$$h_a = c_{pa} \cdot \left(T_{db} - 273.16K \right) \tag{3. 9}$$

where the specific heat capacity of dry air, c_{pa}, is correlated with a third-order polynomial obtained in Excel:

$$c_{pa} = 1.045356 \cdot 10^{3} - 3.161783 \cdot 10^{-1} \cdot (T_{db}) + 7.083814 \cdot 10^{-4} \cdot (T_{db})^{2} - 2.705209 \cdot 10^{-7} \cdot (T_{db})^{3} \quad (3.\ 10)$$

From equations (3.9) and (3.10), the specific enthalpy for dry air at the inlet and outlet can be calculated. The specific enthalpy for water vapour, h_v, is given by the following:

$$h_v = i_{fgwo} + c_{pv} \cdot (T_{db} - 273.16K) \quad (3.\ 11)$$

where the latent heat, i_{fgwo}, is evaluated at 273.15 K and i_{fgwo} = 2.5016x10^{6} J/kg. The specific heat of water vapour is given by:

$$c_{pv} = 1.3605 \cdot 10^{3} + 2.31334 \cdot (T_{db}) - 2.46784 \cdot 10^{-10} \cdot (T_{db})^{5} + 5.91332 \cdot 10^{-13} \cdot (T_{db})^{6} \quad (3.\ 12)$$

To solve equation (3.8), the only value that needs to be calculated is the specific enthalpy of the water liquid that is added to the incoming air, h_f:

$$h_f = c_{pw} \cdot (T_{nb} - 273.16K) + \upsilon_f \cdot P \quad (3.\ 13)$$

where the pressure, P, is assumed to be at atmospheric pressure at sea level and the specific heat of water liquid is given by:

$$c_{pw} = 8.15599 \cdot 10^{3} - 2.8067 \cdot 10 \cdot (T_{nb}) + 5.11283 \cdot 10^{-2} \cdot (T_{nb})^{2} - 2.17582 \cdot 10^{-13} \cdot (T_{nb})^{6} \quad (3.\ 14)$$

and the specific volume:

$$\upsilon_f = 1.49343 \cdot 10^{-3} - 3.7164 \cdot 10^{-6} \cdot (T_{nb}) + 7.09782 \cdot 10^{-9} \cdot (T_{nb})^{2} - 1.90321 \cdot 10^{-20} \cdot (T_{nb})^{6} \quad (3.\ 15)$$

From equations (3.13) to (3.15), the specific enthalpy of the water can be calculated and equation (3.8) can be solved.

In order to calculate the mass transfer of water vapour into the air stream, a semi-empirical equation proposed by Gilliland (Kröger, 1998) was used for calculating the diffusion coefficient:

$$D = 0.04357 \cdot T_{db}^{1.5} \cdot \frac{\left(\frac{1}{M_a} + \frac{1}{M_b}\right)^{0.5}}{P \cdot \left(V_a^{0.333} + V_b^{0.333}\right)^2} \quad (3.\ 16)$$

where V_a and V_b are the molecular volume of gases a and b, which in this case is air and water, and thus, for air $V_a = 29.9$ and for water vapour $V_v = 18.8$. The molecular masses for air and water are $M_a = 28.97$ and $M_v = 18.016$ respectively.

Assuming that the evaporation process of air blown through fries can be approached as the evaporation of liquid inside a circular column, with air blown through it and diameter, *d*, Gilliland (Kröger, 1998) supplied the following semi-empirical equation:

$$Sh = \frac{h_D \cdot d}{D} = 0.023 \cdot \mathrm{Re}^{0.83} \cdot Sc^{0.44} \quad (3.\ 17)$$

valid for $2000 < Re < 35000$ and $0.6 < Sc < 2.5$. The Schmidt number is given by:

$$Sc = \frac{\mu}{\rho \cdot D} \quad (3.\ 18)$$

and the Reynolds number is given by:

$$\mathrm{Re} = \frac{\rho \cdot V \cdot d}{\mu} = \frac{G \cdot d}{\mu} \quad (3.\ 19)$$

and the mass flux, G, is given by:

$$G = \frac{m_{air}}{A_{columns}} = \frac{\rho \cdot V \cdot A_{air}}{A_{columns}} \qquad (3.20)$$

To solve equation (3.17), equations (3.16) to (3.20) were set into equation (3.17). The relation between the area of the fries and the flow-through area of the columns between the fries was approximated as $A_{air}/A_{columns}$ = 20/1 and the diameter of the columns through the fries approximated as d = 0.003 m:

$$h_D = 0.742 \cdot \left(\frac{\rho}{\mu}\right)^{0.39} \cdot V^{0.83} \cdot D^{0.56} \qquad (3.21)$$

The mass transfer coefficient is:

$$h_d = \frac{h_D \cdot P}{R_v \cdot (\omega_{sat} - \omega)} \cdot \left[\frac{\omega_{sat}}{T_f \cdot (\omega_{sat} + 0.622)} - \frac{\omega}{T_{db}(\omega + 0.622)}\right] \qquad (3.22)$$

From equation (3.22), it is possible to calculate the mass transfer rate:

$$m_w = h_d \cdot A \cdot (\omega_{sat} - \omega) \qquad (3.23)$$

Figures 18 to 20 were generated from the equations above. The calculations were done in Excel (see appendix J.2).

3.6.2 Boiler

Boiler energy losses which occur are radiation losses, exhaust gas losses, poor boiler management and combustion inefficiencies (these include oil burner inefficiencies and non-stoichiometric combustion). Boiler efficiency can be improved through improved combustion, flash-steam heat recovery, flue-gas heat recapturing (recuperator and economiser) and improved TDS (Total Dissolved Solids) management. Currently, Lambert's Bay has a coal-fire tube boiler with a steam flow rate of 10 ton per hour (10 000 kg/h) and a design steam pressure of 22 bar (2.2 MPa). The boiler lacks an economiser, automatic TDS control, separator or superheating of steam, preheating of air and blow-down heat recovery. The addition of these features will be discussed in the following section.

Economisers reduce the heat in the flue gas and transfer it to the boiler feed water. They can increase the efficiency of boilers by 3% – 5% (Dutch National Team, 2001). Oxygen trim controllers control the fuel-to-air ratio within optimal limits. An oxygen trim control system will correct the airflow so that the combustion efficiency remains as high as possible. It can improve boiler efficiency by 1% – 2%. Blow-down heat recovery (recovers heat and water from TDS blow-down) recovers the flash steam resulting from boiler blow-down to a lower pressure. Steam boilers need to be blown down in order to control the TDS level in the boiler water between acceptable limits. This is best achieved by using a TDS control system, which will open a valve to allow boiler water to discharge when the TDS level rises above a pre-set limit. Relatively low TDS feed water then replaces the discharged boiler water (up to 80% of the heat in the discharge is recoverable by using properly designed flash vessels and heat exchangers). Due to the abnormally high TDS levels (between 700 ppm and 1100 ppm) for Lambert's Bay's municipal water, this document will look at energy savings through flash-steam heat recovery.

Flash-steam Heat Recovery

Flash steam is released from the hot blow-down water when the pressure drops as shown in figure 22. This mixture of flash steam and blow-down water is allowed to separate in a flash vessel (the design of the flash vessel ensures that flow velocities are low for promoting good separation). The dry flash steam is then introduced at low pressure to the feed-water storage tank through the steam distributor or flash condensing deaerator head. A float trap fitted to the outlet of the flash vessel drains the residual blow-down. From the float trap, the residual blow-down water is allowed to pass into the heat exchanger, where it gives up its heat to circulating cold make-up water. The cooled blow-down water then flows safely to the drain. To save energy, the circulating pump is only activated by a thermostat mounted on the blow-down water inlet to the heat exchanger. Therefore, it will only run when blow-down water is flowing.

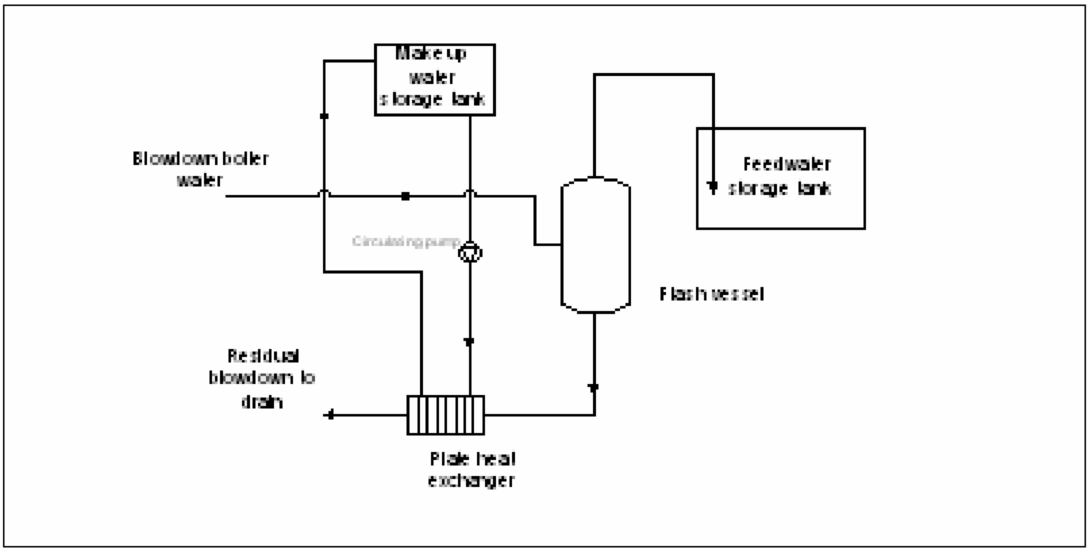

Figure 22: Flash-steam heat recovery system with plate heat exchanger

TDS levels for boilers need to be less than 2500 ppm to 3000 ppm. Through better treatment of the feed water, TDS levels can be decreased, thus leading to less blow-down.

Assuming a relatively constant boiler blow-down rate when operating at maximum demand with the feed-water TDS level at 700 ppm and the boiler TDS level at 3000 ppm, the blow-down rate can be calculated:

$$\dot{m}_{fw} \cdot ppm_{fw} = \dot{m}_{bd} \cdot ppm_{boiler} \tag{3. 24}$$

where m_{fw} is the feed-water flow rate. The feed-water flow rate is 10 000 kg/h and thus the blow-down rate can be calculated:

$$\dot{m}_{bd} = \frac{ppm_{fw}}{ppm_{boiler}} \cdot \dot{m}_{fw} \tag{3. 25}$$

The percentage of flash steam recovered in the flash vessel can be calculated through an energy balance, assuming that the blow-down condensate flashes off to 0.2 MPa (2 bar guage, 3 bar absolute pressure):

$$\begin{aligned} \dot{m}_{bd} \cdot h_{f\,23bar} &= \dot{m}_{flash} \cdot h_{g\,3bar} + \left(\dot{m}_{bd} - \dot{m}_{flash}\right) \cdot h_{f\,3bar} \\ \dot{m}_{flash} &= \frac{h_{f\,23bar} - h_{f\,3bar}}{h_{g\,3bar} - h_{f\,3bar}} \cdot \dot{m}_{bd} \end{aligned} \tag{3. 26}$$

Thus, the rate of energy recovered by flash-steam heat recovery is:

$$\dot{E}_{flashrec} = \dot{m}_{flash} \cdot h_{g3bar} \tag{3. 27}$$

By substituting $\dot{m}_{flash}$ into the above equation and replacing $\dot{m}_{bd}$, the recovered flash steam is:

$$\dot{E}_{flashrec} = \frac{h_{f23bar} - h_{f3bar}}{h_{g3bar} - h_{f3bar}} \cdot \frac{ppm_{fw}}{ppm_{boiler}} \cdot \dot{m}_{fw} \cdot h_{g3bar} \tag{3. 28}$$

Substituting the variables in the above equation:

$$\begin{aligned} \dot{E}_{flashrec} &= \frac{941.614kJ/kg - 561.47kJ/kg}{2725.3kJ/kg - 561.47kJ/kg} \cdot \frac{700\,ppm}{3000\,ppm} \cdot 2.7778kg/s \cdot 2725.3kJ/kg \\ \dot{E}_{flashrec} &= 310kW \end{aligned} \tag{3. 29}$$

This figure of 310 kW is obtained without further heat recovery of the condensate. The flash steam that is recovered also leads to an improved TDS value and less oxygen in the feed water. The equipment necessary to obtain these savings amount to R44 206,00 (Spirax-Sarco quote reference nr. NL5287S). The monthly energy savings for the flash heat recovery system is given by:

$$E_{SM} = \frac{\eta_{avail} \cdot C_{coal}}{\eta_{conversion} \cdot Cal_{coal}} \cdot t_{month} \cdot \dot{E}_{flashrec} \tag{3. 30}$$

With the availability η_{avail} = 0.97, the conversion efficiency from coal to steam $\eta_{conversion}$ = 0.85, the calorific value of coal Cal_{coal} = *27 MJ/kg*, the cost of coal C_{coal} = R400/ton, and the time in a month, t_{month} = 2.628 x 10^6 s, the energy savings per month are:

$$E_{SM} = R13773.24 \tag{3. 31}$$

The payback period is given by equation (3.2), with P the cost of system and A the monthly energy savings:

$$n = \frac{\ln\left(\dfrac{1}{1-\dfrac{P}{A}\cdot i}\right)}{\ln(1+i)} \tag{3. 32}$$

Working with an average interest rate of 17% per annum (1.41667% per month), the payback period can be calculated:

$$n = \frac{\ln\left(\dfrac{1}{1-\dfrac{44206}{13773.24}\cdot 0.0141667}\right)}{\ln(1+0.0141667)} = 3.308 \qquad \text{months} \tag{3. 33}$$

Thus, the flash-steam heat recovery system as proposed above has a payback period of 3.3 months. In calculating the payback period, the installation cost and extra maintenance cost were ignored.

3.6.3 Utilisation of Reject Steam at Steam Peeler

The steam peeler's function is to remove the potato skin at the required flow rate. The steam peeler removes the skin of the potatoes by means of a rapid pressure release on the raw potatoes. This steam-potato mixture is then freely ejected into the atmosphere, where it is separated in a deceleration duct and the steam goes upward and the skin slurry into bins which are discarded. Figure 23 illustrates the expanded steam from the steam peeler that condenses in the atmosphere without any recapturing of energy.

Figure 23: Rejected steam of steam peeler into atmosphere

Description and Mechanism of Steam Peeler

The steam peeler consists of a peeling vessel and a weighing hopper, which weighs a specific amount of product with each batch of potatoes. When the product has entered the steam peeler, the peeling vessel cover closes and steam is supplied through the central steam distribution grill into the vessel at approximately 1.6 MPa (16 bar) absolute pressure. The steam is equally divided and mixed through the product. The Dutch company Uticon-Dynatherm Industrie bv estimates that 14% of a potato's volume is lost in a steam-peeling action in their energy calculations for the Lambert's Bay Canning company. It will be a safe approximation to assume that 14% of the potatoes' volume that is lost contributes to the same volume of steam being lost for any further heat recovery. A certain amount of condensate is lost for further use and a small amount of energy is used as sensible heat added to the potatoes, but these can all be assumed to be negligible compared to the volume of steam assumed to be lost for any further heat recovery. To estimate the mass flow rate of the product at different stages on the production line, a percentage loss diagram is shown for each subsystem in the production line (table 3.1). These values were obtained from the feasibility study Uticon-Dynatherm Industrie bv did before erecting the plant. Calculations were done for a final product output of 100 ton per day with an availability of 0.97. Thus, the mass flow of the final product out:

$$\dot{m}_{product,out} = \frac{100 \cdot 1000}{24 \cdot 0.97} = 4295.5 kg/h \qquad (3.\ 34)$$

Mass flow	% loss of total input	Process	% loss at process
9406.637	100.0%	Input	
9406.637	100.0%	Peeling	14.003%
8089.426	86.0%	Cutting/sorting/grading	15.044%
6872.453	73.1%	Blanching	2.083%
6729.299	71.5%	Drying	24.890%
5054.377	53.7%	Frying	12.465%
4424.349	47.0%	Defatting	0.971%
4381.388	46.6%	Cooling and freezing	0.980%
4338.451	46.1%	Packing	0.990%
4295.5	45.7%	Output	

Table 3. 1: Percentage loss of product mass

For a production rate of 1 000 kg/h of raw potatoes at the steam peeler, 100 kg/h of saturated steam at 1.6 MPa (16 bar) absolute pressure is required (Uticon-Dynatherm Industrie bv, 1994). The enthalpy of saturated steam at 16 bar is 2793.88 kJ/kg (Çengel & Boles, 1994). With an availability of 0.97 and a daily production rate of 100 000 kg/day, the total amount of steam that enters the steam peeler is:

$$\dot{m}_{steam,in} = \frac{0.1 \cdot 100.0\%}{45.7\%} \cdot \dot{m}_{product,out} = 0.21882 \cdot \dot{m}_{product,out} \qquad (3.\ 35)$$

The amount of energy in the steam that enters the steam peeler is:

$$\dot{E}_{steam,in} = \dot{m}_{steam,in} \cdot h_{g,16bar} \qquad (3.\ 36)$$

Figure 24 below shows the energy flow during one full cycle of the steam peeler.

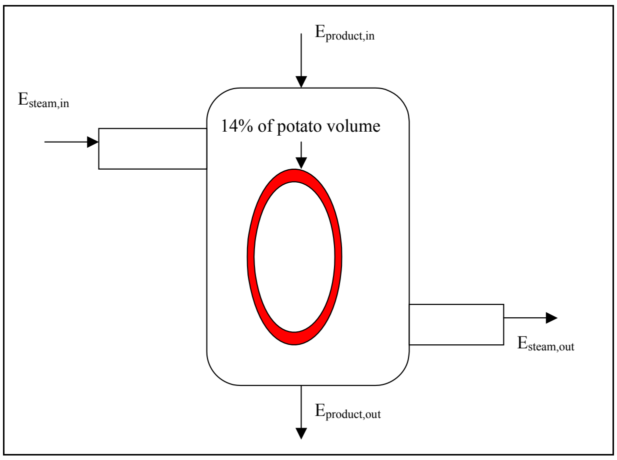

Figure 24: Steam peeler energy flow

The steam energy available for utilisation is:

$$\dot{E}_{steam,out} = \dot{E}_{steam,in} + \dot{E}_{product,in} - \dot{E}_{product,out} \tag{3. 37}$$

where the energy added to the raw potatoes is approximated as the volume of steam required to equal 14% of the total potato. The amount of steam is then:

$$\dot{m}_{loss,steam} = \frac{\dot{V}_{loss,potato}}{v_{g,16bar}} = \frac{0.14 \cdot \dot{m}_{potato}}{\rho_{potato} \cdot v_{g,16bar}} \tag{3. 38}$$

where the potato density, ρ_{potato}, is 1075 kg/m^3 (Garrote, Silva & Bertone, 2000) and the mass flow of the potato is taken as 100 ton per day with an availability of 0.97, and the specific volume of steam at 1.6 MPa (16 bar) absolute pressure, $v_{g,16bar}$, is 0.124458 m^3/kg (Çengel & Boles, 1994). The energy flow rate of the steam that is wasted is then:

$$\begin{aligned}
&\dot{E}_{steam,out} = \left(\dot{m}_{steam,in} - \dot{m}_{loss,steam}\right) \cdot h_{g,16bar} \\
&\therefore = \left(0.21882 - \frac{0.14}{\rho_{potato} \cdot \upsilon_{g,16bar}}\right) \cdot \dot{m}_{product,out} \cdot h_{g,16bar} \\
&\therefore = 700.64 kW
\end{aligned} \tag{3. 39}$$

Thus, for an output of 4295.5 kg/h of French fries, 700.64 kW energy in the form of steam is discarded into the environment. With the assumption that the pressure inside the vessel is approximately 1.6 MPa (16 bar) absolute pressure, and at saturation, this energy could be used as an energy source for the hot water system and the defrosting system, and the rest could be used as energy to preheat fresh water for the blanchers. Or, to minimise capital expenditure all of the heat can be used to preheat the fresh water for the blanchers. The schematic representation in figure 25 shows the current situation at the steam peeler.

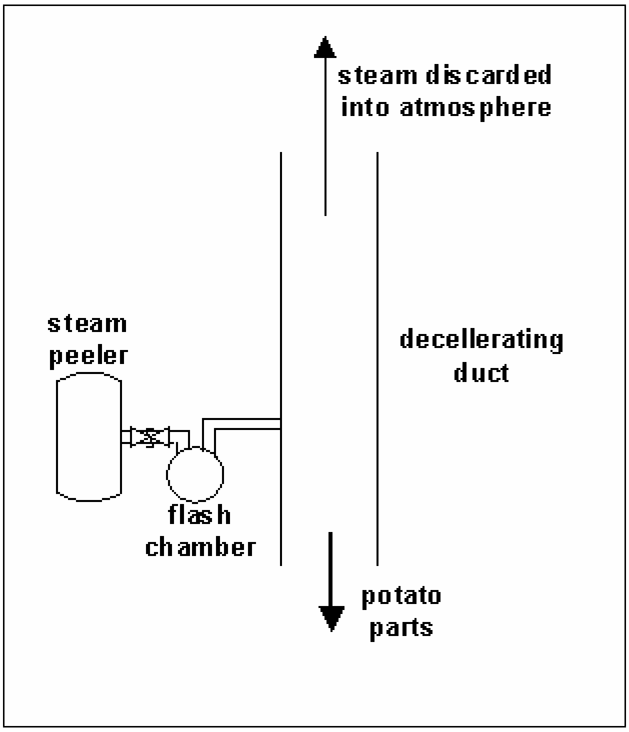

Figure 25: Schematic representation of steam peeler current operation

A concept proposal (see figure 26) for the utilisation of the waste steam was developed and payback periods were calculated for different heat exchangers. In this concept, the flash chamber was discarded and a protection plate was placed on the inside of the deceleration duct. This concept makes use of a finned-tube heat exchanger that transfers heat to the process water of the blanchers.

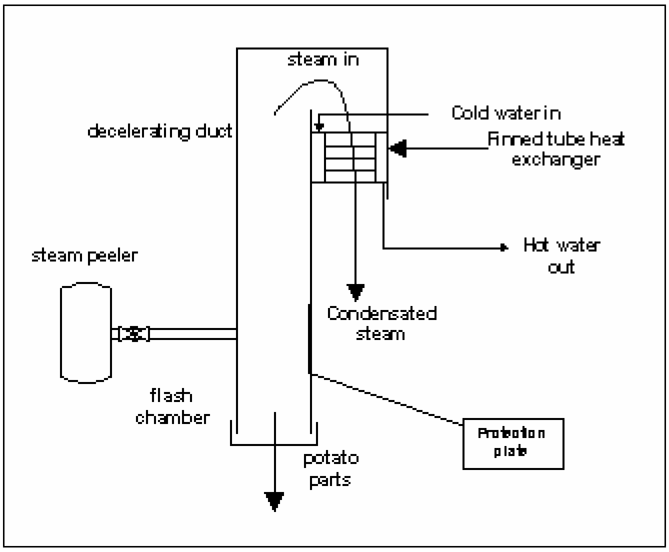

Figure 26: Steam peeler with shell-tube heat exchanger

Energy Transfer Analysis of Finned-tube Heat Exchanger

A computer program was written, using Microsoft Visual Basic V.6 to evaluate a finned tube configuration for the utilisation of reject steam (appendix K). A control volume was set up and heat transfer and pressure drop calculations were done for the control volume (see figure below for the control volume boundaries).

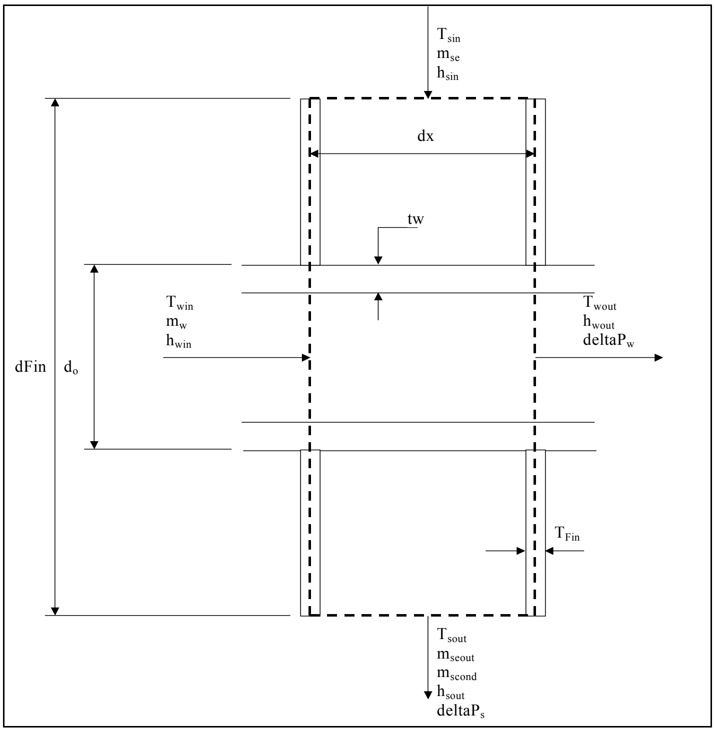

Figure 27: Control volume for finned tube

The overall heat transfer coefficient, U_o, is based on the outside surface of the tube and was determined for the control volume in order to determine the increase of the water enthalpy and the decrease of the steam enthalpy. The overall heat transfer coefficient is given by the following equation (Kröger, 1998):

$$U_o = \left[\frac{A_o}{A_i \cdot h_i} + \frac{A_o \cdot \ln\left(\frac{r_o}{r_i}\right)}{2 \cdot \pi \cdot k \cdot L} + \frac{1}{h_o \cdot e_f} \right]^{-1} \qquad (3.\ 40)$$

where A_o and A_i is the respective outside and inside surface areas, h_i the inner convective heat transfer coefficient – in this case for the water, r_o and r_i the respective outer and inner radii of the tube, k the thermal conductivity of the tube material, L the length of the control volume considered – in this case dx, h_o the outside convective heat transfer coefficient and e_f the fin effectiveness.

The outside convective heat transfer coefficient for the steam was calculated at T_{sin}, the steam temperature. Correlations (see appendix K.3) obtained in Excel for the density, ρ_s, kinematic viscosity, υ_s, conduction coefficient, k_s, and the Prandtl number, Pr_s, for the steam were set up in the heat exchanger program code. The abovementioned properties were obtained from Schmidt (1969).

The Reynolds number is given by:

$$\mathrm{Re}_{ds} = \frac{m_{se} \cdot d_o}{\rho_s \cdot A_{fr} \cdot \upsilon} \tag{3. 41}$$

with m_{se} the mass flow of the steam for each element and A_{fr} the frontal area perpendicular to the steam flow. The average Nusselt number for Pr > 0.5 is given by the correlation suggested by Churchill and Bernstein (Mills, 1995). For Re_{ds} < 10 000 the Nusselt number is:

$$Nu_{ds} = 0.3 + \frac{0.62 \cdot \mathrm{Re}_{ds}^{1/2} \cdot \mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}} \tag{3. 42}$$

and for 10 000 < Re_{ds} < 4 x 10^5:

$$Nu_{ds} = 0.3 + \frac{0.62 \cdot \mathrm{Re}_{ds}^{1/2} \cdot \mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}} \cdot \left[1 + \left(\frac{\mathrm{Re}_{ds}}{282000}\right)^{1/2}\right] \tag{3. 43}$$

and for 4 x 10^5 < Re_{ds} < 5 x 10^6:

$$Nu_{ds} = 0.3 + \frac{0.62 \cdot \mathrm{Re}_{ds}^{1/2} \cdot \mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}} \cdot \left[1 + \left(\frac{\mathrm{Re}_{ds}}{282000}\right)^{5/8}\right]^{4/5} \tag{3. 44}$$

With the Nusselt number known, the outside convective heat transfer coefficient for the steam, h_{cs}, can be calculated:

$$h_{cs} = \left(\frac{k_s}{d_o}\right) \cdot Nu_{ds} \tag{3. 45}$$

For the calculation of the convective heat transfer coefficient for the water, the same approach was followed. Correlations were set up in Excel for the density, ρ_w, kinematic viscosity, υ_w, conduction coefficient, k_w, and the Prandtl number, Pr_w, of the water (see appendix K.3). The Reynolds number for the water side is given by:

$$\mathrm{Re}_{dw} = \frac{m_w \cdot d_i}{\rho_w \cdot A_{di} \cdot \upsilon_w} \tag{3. 46}$$

For turbulent flow, the wall friction is needed in order to calculate the convective heat transfer coefficient for the water. The friction factor for a smooth wall from Petukhov's formula (Mills, 1995) is:

$$f = (0.790 \cdot \ln \mathrm{Re}_{dw} - 1.64)^{-2} \tag{3. 47}$$

For $\mathrm{Re}_{dw} < 3000$, the flow is laminar and the Nusselt number is given by:

$$Nu_{dw} = 4.364 \tag{3. 48}$$

For $3000 < \mathrm{Re}_{dw} < 10\,000$, Gnielinski's formula is recommended (Mills, 1995):

$$Nu_{dw} = \frac{(f/8) \cdot (\mathrm{Re}_{dw} - 1000) \cdot \mathrm{Pr}_w}{1 + 12.7 \cdot (f/8)^{1/2} \cdot (\mathrm{Pr}^{2/3} - 1)} \tag{3. 49}$$

For $\mathrm{Re}_{dw} > 10\,000$, a simple power law formula is used:

$$Nu_{dw} = 0.023 \cdot \mathrm{Re}_{dw}^{0.8} \cdot \mathrm{Pr}^{0.4} \tag{3. 50}$$

From the above correlations, the convective heat transfer coefficient for the water is calculated as follows:

$$h_{cw} = \left(\frac{k_w}{d_i}\right) \cdot Nu_{dw} \tag{3. 51}$$

The fin effectiveness is represented by the following equation (Kröger, 1998):

$$\eta_{fin} = \frac{\tanh\left(b_{fin} \cdot d_o \cdot \phi_{fin} / 2\right)}{b_{fin} \cdot d_o \cdot \phi_{fin} / 2} \tag{3. 52}$$

where the above gives the fin efficiency with:

$$b_{fin} = \frac{h_{cs}}{\frac{t_{fin} \cdot k_{fin}}{2}} \tag{3. 53}$$

and

$$\phi_{fin} = \left(\frac{r_{2fin}}{r_{1fin}} - 1\right) \cdot \left(1 + 0.35 \cdot \ln\left(\frac{r_{2fin}}{r_{1fin}}\right)\right) \tag{3. 54}$$

where r_{2fin} and r_{1fin} is the outer and inner radii of the annular fin. To calculate the hyperbolic tan in the program, the mathematical relations below were used. The hyperbolic tan of any function z is given by (Greenberg, 1998):

$$\begin{aligned} \tanh(z) &= \frac{\sinh(z)}{\cosh(z)} \\ \tanh(z) &= \frac{e^{z} - e^{-z}}{e^{z} + e^{-z}} \\ \tanh(z) &= \frac{1 - e^{-2z}}{1 + e^{-2z}} \end{aligned} \tag{3. 55}$$

where e^{-2z} is approximated by the first six terms in the following Taylor series (Greenberg, 1998):

$$e^{-2z} = 1 + (-2z) + \frac{(-2z)^2}{2!} + \frac{(-2z)^3}{3!} + \frac{(-2z)^4}{4!} + \frac{(-2z)^5}{5!} \tag{3. 56}$$

where $z = b_{fin} \cdot d_o \cdot \phi_{fin} / 2$.

The fin effectiveness can now be calculated:

$$e_{fin} = 1 - \frac{\left(1 - \eta_{fin}\right) \cdot A_{fin}}{A_{total}} \tag{3. 57}$$

where A_{fin} is the outside fin area and A_{total} is the total outside fin and tube area. With the abovementioned calculated for the control volume, the outlet enthalpies for the steam and water, H_{sout} and H_{wout}, can be calculated:

$$\begin{aligned} H_{sout} &= H_{\sin} - \Delta Q_{cv} \\ H_{wout} &= H_{win} + \Delta Q_{cv} \end{aligned} \tag{3. 58}$$

The enthalpy values of steam and water were obtained from Schmidt (1969) and are approximated by the following linear equations respectively:

$$\begin{aligned} H_s &= m_s \cdot h_s + m_{sc} \cdot h_{fs} \\ \therefore &= m_s \cdot \left(1986.3 \cdot T_s + 1935600\right) + m_{sc} \cdot \left(419040\right) \end{aligned} \tag{3. 59}$$

$$H_w = m_w \cdot h_w = m_w \cdot \left(4185.1 \cdot T_w + 1143000\right) \tag{3. 60}$$

The unit for H_s and H_w is *Watt*, and m_s and m_w are the steam and water mass flow rates respectively, m_{sc} is the condensate mass flow rate and h_{fs} the enthalpy of saturated liquid water at 1 atm pressure. Condensation will start at 373.15 K (100 ^oC) and from there the steam flow rate will decrease with the condensate flow rate given by:

$$m_{sc} = \frac{\Delta Q_{cv}}{h_{fgs}} \tag{3. 61}$$

with h_{fgs} the specific enthalpy of evaporation at 1 atm absolute pressure and Q_{cv} the heat transfer to the water. Although the pressure in the heat exchanger will be slightly higher than atmospheric, the steam can be regarded as if it is at atmospheric pressure.

For the pressure drop on the outside of the tubes, Zukauskus (Mills, 1995) recommends the following:

$$\Delta P_s = N \cdot \chi \cdot \left(\frac{\rho \cdot V_{\max}^2}{2} \right) \cdot f \tag{3. 62}$$

where the friction factor f and correction factor χ are calculated for the worst conditions in the heat exchanger and N is the number of tube banks. The transverse pitch, $S_{T,}$ is 0.05 m and the longitudinal pitch, S_L, is 0.08 m. With an outside tube diameter of 0.025 m, the dimensionless transverse pitch, P_T, and the dimensionless longitudinal pitch, P_L, are (Mills, 1995):

$$\begin{aligned} P_T &= S_T / d = 2 \\ P_L &= S_L / d = 3.2 \\ P_T / P_L &= 0.625 \end{aligned} \tag{3. 63}$$

V_{max} is the maximum velocity in the tube bank and for an aligned bank, the maximum velocity occurs between adjacent tubes of a transverse row:

$$V_{\max} = V_o \cdot \frac{S_T}{S_T - D} \tag{3. 64}$$

The pressure drop inside the water tubes was calculated for a tube with a relative roughness factor of $\varepsilon/d = 0.00006$. For $\varepsilon/d > 10^{-4}$, Haaland (Kröger, 1998) recommends the following equation for the friction factor:

$$f_w = 0.3086 \cdot \left(\frac{\ln\left(\frac{6.9}{\mathrm{Re}_w} + \left(\frac{\varepsilon / d}{3.75} \right)^{1.11} \right)}{\ln 10} \right)^{-2} \qquad (3.\ 65)$$

and for a very small relative roughness, ε/d, the friction factor is given by:

$$f_w = 2.7778 \cdot \left(\frac{\ln\left(\left(\frac{7.7}{\mathrm{Re}_w} \right)^{3} + \left(\frac{\varepsilon / d}{3.75} \right)^{3.33} \right)}{\ln 10} \right)^{-2} \qquad (3.\ 66)$$

The pressure drop over the control volume for the water is given by:

$$\Delta P_w = \frac{f_w \cdot dx^2 \cdot m_w^2 \cdot}{d_i \cdot \rho_w \cdot A_i^2} \qquad (3.\ 67)$$

A routine (appendix K) was written for the energy balance of the segment in Visual Basic (programming language) to calculate the overall performance of the proposed heat exchanger.

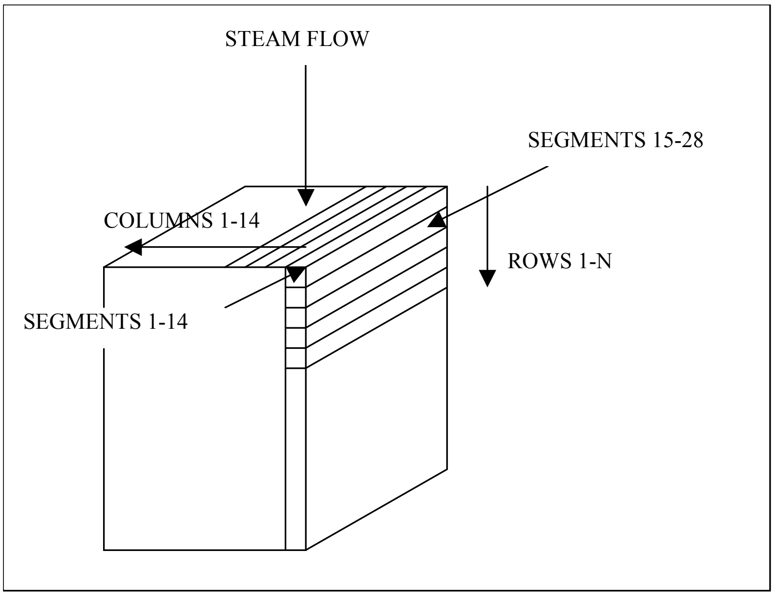

Figure 28: Tube bank configuration of heat exchanger

This routine was run for a square, aligned tube-bank configuration heat exchanger with fins, without fins and for a combination of fins and finless tubes, and the results are shown in figures 28 and 29 and tables 3.2 and 3.3. The simulation was run for one tube (one column), with the assumption of uniform flow over all 14 tubes (see figure 28).

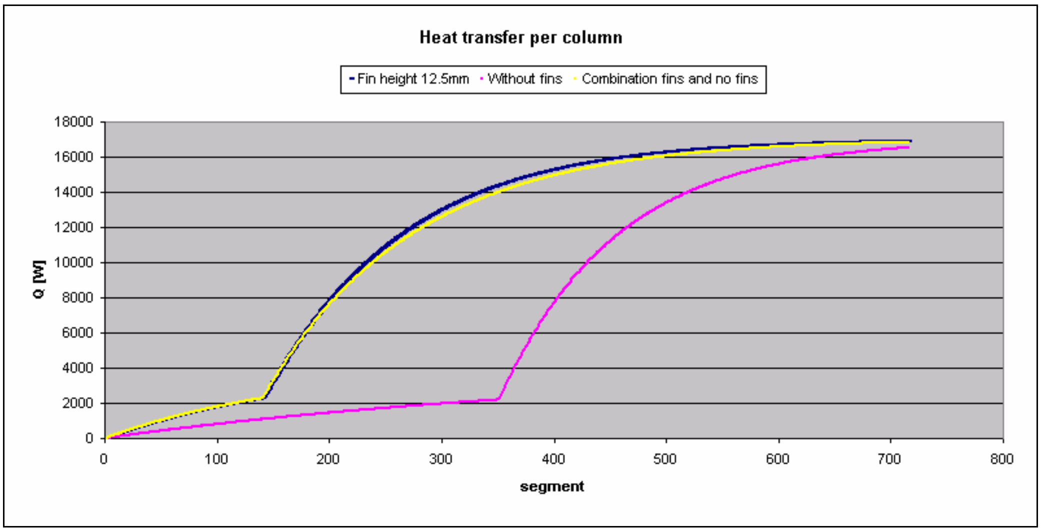

Figure 29: Heat transfer for rectangular heat exchanger concept

In figure 29 it is clear that above 700 segments per tube the increase in heat load, Q, will be insignificant. With 14 segments per row this implies 50 rows per tube. From figure 30, for the best curves one can see that above 500 segments the increase in the heat load is insignificant. Thus for more than 35 rows per tube, the increase in heat load is insignificant.

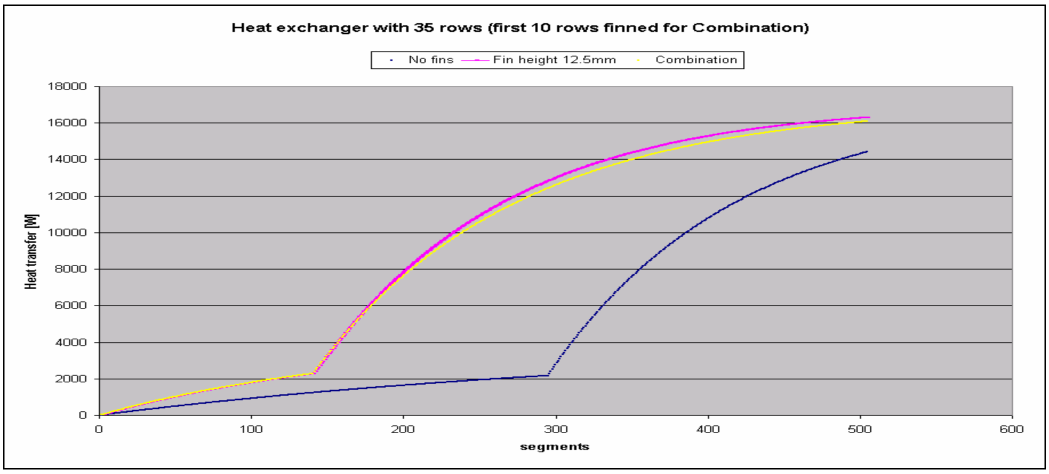

Figure 30: Heat transfer for rectangular heat exchanger with 35 layers

Figure 30 shows the heat load increase until 500 segments (35 rows). Results obtained from the program for finned tubes, finless tubes and a combination of finned and finless tubes are tabulated below. 1 column represents results for one tube of the heat exchanger evaluated for all the rows. The heat exchanger is composed of 14 columns (given there are 14 tubes parallel to each other).

Heat exchanger simulation results (50 layers)				
	Fin height 12.5 mm		No fins	
	1 Column	Heat exchanger	1 Column	Heat exchanger
Steam pressure fall	261.39 Pa	261.39 Pa	321.89 Pa	321.89 Pa
Water pressure fall	389 484.62 Pa	389 484.62 Pa	423 452.10 Pa	423 452.10 Pa
Condensate flow	6.4766×10^{-3} kg/s	9.0672×10^{-2} kg/s	6.387335×10^{-3} kg/s	8.9423×10^{-2} kg/s
Steam flow	1.20801×10^{-2} kg/s	1.6912×10^{-1} kg/s	1.21874×10^{-2} kg/s	1.70624×10^{-1} kg/s
Heat transferred	16 903.7 W	236.6518 kW	16 584.22 W	232.17908 kW
Heat transfer surface area	7.0096786 m^2	98.1355 m^2	2.80387 m^2	39.254 m^2
	Combination of fins and no fins			
Steam pressure fall	264.26 Pa	264.26 Pa		
Water pressure fall	391599.45 Pa	391599.45 Pa		
Condensate flow	6.4505×10^{-3} kg/s	9.0307×10^{-2} kg/s		
Steam flow	1.210877×10^{-2} kg/s	1.6952×10^{-1} kg/s		
Heat transferred	16 844.78 W	235.8269 kW		
Heat transfer surface area	3.62854 m^2	50.7995 m^2		

Table 3. 2: Heat exchanger simulation results for 50 layers

Heat exchanger simulation results (35 layers)				
	Fin height 12.5 mm		No fins	
	1 Column	Heat exchanger	1 Column	Heat exchanger
Steam pressure fall	152.05 Pa	152.05 Pa	205.52 Pa	205.52 Pa
Water pressure fall	287 911.55 Pa	287 911.55 Pa	317 387.99 Pa	317 387.99 Pa
Condensate flow	6.1752×10^{-3} kg/s	8.6453×10^{-2} kg/s	5.2923×10^{-3} kg/s	7.4092×10^{-2} kg/s
Steam flow	1.23778×10^{-2} kg/s	1.7329×10^{-1} kg/s	1.326857×10^{-2} kg/s	1.8576×10^{-1} kg/s
Heat transferred	16 223.42 W	227.1279 kW	14 112.74 W	197.578 kW
Heat transfer surface area	4.9480084 m^2	69.2721 m^2	1.9792 m^2	27.709 m^2
	Combination of fins and no fins			
Steam pressure fall	154.23 Pa	154.23 Pa		
Water pressure fall	289 302.87 Pa	289 302.87 Pa		
Condensate flow	6.0746×10^{-3} kg/s	8.5044×10^{-2} kg/s		
Steam flow	1.2479×10^{-2} kg/s	1.7471×10^{-2} kg/s		
Heat transferred	15 996.39 W	223.949 kW		
Heat transfer surface area	2.80387 m^2	39.2542 m^2		

Table 3. 3: Heat exchanger simulation results for 35 layers

From equation (3.2) the payback period for each different heat exchanger configuration is obtained by assuming a cost per m^2, C_A. With the total heat transfer area, A_t greater than 5 m^2, the cost per m^2, C_A, as a rule of thumb can be approximated as R3 000/m^2 to R4 000/m^2. Assuming the monthly payment, A, to be equal to the monthly energy savings, E_{SM}, the number of periods is given by the following equation:

$$n = \frac{\ln\left(\dfrac{1}{1 - \dfrac{C_A \cdot A_t}{E_{SM}} \cdot i}\right)}{\ln(1+i)} \qquad (3.\ 68)$$

where the monthly interest rate, i, equals the average prime rate for South Africa over the last five years. The monthly energy savings are:

$$E_{SM} = \frac{\eta_{avail} \cdot C_{coal}}{\eta_{conversion} \cdot Cal_{coal}} \cdot t_{month} \cdot Q_{transfer} \qquad (3.\ 69)$$

where the availability of the plant, η_{avail}, is 0.97, the conversion efficiency from coal to steam, $\eta_{conversion}$, is 0.85, the cost of coal, C_{coal}, is R400/ton and the amount of seconds in a month, t_{month}, is 2.628 x 10^6 s. Table 3.4 shows the different payback periods calculated for the different heat exchanger configurations:

Heat exchangers with 50 tube layers (longitudinal direction)			
	finned	finless	combination
Heat exchanger cost estimate	R392 542	R157 016	R203 198
Monthly energy savings	R10 391.24	R10 194.84	R10 355.02
Payback period	54.46 months	17.50 months	23.15 months
Heat exchangers with 35 tube layers (longitudinal direction)			
	finned	finless	combination
Heat exchanger cost estimate	R277 088	R110 836	R157 017
Monthly energy savings	R9 973.05	R8 675.53	R9 833.47
Payback period	35.56 months	14.19 months	18.23 months

Table 3. 4: Payback periods for heat exchanger configuration

From table 3.4, the finless heat exchanger offers the shortest payback period and would be the logical choice when capital expenditure is a limiting variable.

3.6.4 Steam Pipes

No significant signs of heat wastage due to uninsulated or damaged pipes were found. Most of the pipes were insulated, although some of the insulation was slightly damaged during maintenance procedures. Calculations (appendix D) show that an uninsulated 50-mm diameter pipe of 1 m length and operating at 0.8 MPa (8 bar) guage pressure can cost the plant approximately R215 per annum.

Due to the plant being equipped with so few unlagged steam pipes, measurements for the steam pipes were not taken, but calculations were conducted for 1-m lengths of unlagged piping – either horizontal or vertical. Two heat transfer models will be used for the steam pipes: natural convection over a horizontal cylinder and natural convection over a wall for a vertical pipe. To simplify calculations, the unlagged piping must be divided into segments.

Sample calculations (appendix D) were done to estimate the heat loss due to natural convection into the plant environment. Horizontal and vertical pipes were considered, but it was found that convection heat loss from vertical and horizontal piping is almost the same.

Approach for Calculation of Heat Loss for Uninsulated Pipes

For natural convection over a horizontal cylinder, the Grashof, Nusselt and Rayleigh numbers were calculated with:

$$\beta = \frac{1}{T_{dbplant}} \tag{3. 70}$$

$$k = 0.026 \frac{W}{m \cdot K} \tag{3. 71}$$

$$\Pr = 0.69 \tag{3. 72}$$

$$\upsilon = 14.7 \cdot 10^{-6} \frac{m^2}{s} \tag{3. 73}$$

$$g = 9.807 \frac{m}{s^2} \tag{3. 74}$$

where β is the volumetric coefficient of expansion, k the conductivity, Pr the Prandtl number and ν the kinematic viscosity coefficient for air (these values were all calculated at known atmospheric conditions).

The Grashof and Rayleigh numbers are given by the following equations respectively:

$$Gr_{Di} = \frac{\left(\beta \cdot \Delta T_i \cdot g \cdot d_i^3\right)}{\nu^2} \tag{3. 75}$$

$$Ra_{Di} = Gr_{Di} \cdot \Pr \tag{3. 76}$$

Where $\Delta T_i = T_i - T_{dbplant}$, the difference between the pipe surface temperature and plant dry-bulb temperature is T_i and $T_{dbplant}$ respectively and g is the gravitational constant. For $Ra_{Di} < 10^9$ laminar flow occurs, but at $Ra_{Di} \sim 10^9$ transition to turbulent flow occurs. Thus, the laminar Nusselt number and turbulent Nusselt number are given by the following equations respectively (Mills, 1995):

$$Nu_{Dlam} = 0.36 + \frac{\left(0.518 \cdot Ra_{Di}^{0.25}\right)}{\left[1 + \left(\frac{0.559}{\Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}} \quad \text{for } Ra_{Di} < 10^9 \tag{3. 77}$$

$$Nu_{Dtur} = \left[0.60 + 0.387 \cdot \left[\frac{Ra_{Di}}{\left[1 + \left(\frac{0.559}{\Pr}\right)^{\frac{9}{16}}\right]^{\frac{16}{9}}}\right]^{\frac{1}{6}}\right] \tag{3. 78}$$

for $Ra_{Di} > 10^9$(3.11)

With the Nusselt number known (laminar or turbulent), the convective heat transfer coefficient for natural convection is:

$$h_{ci} = k \cdot \frac{Nu_{Di}}{d_i} \tag{3. 79}$$

and the heat loss for the uninsulated pipe is calculated as follows:

$$Q_{coni} = h_{ci} \cdot A_{di} \cdot \Delta T_i \tag{3. 80}$$

where A_{di} is the pipe area that is exposed to the plant environment. For radiation heat transfer, the hemispherical emittance, ε_s, is 0.20, and the solar radiation absorptance, α_{ss}, is 0.89, for blue stainless-steel pipes were used. The sky emissivity (see equation 3.89), ε_{sky}, was calculated (Mills, 1995) with typical dry and wet-bulb temperatures inside the plant. Sky irradiation is given by:

$$G_{sky} = \varepsilon_{sky} \cdot \sigma \cdot T_{dbplant}^4 \tag{3. 81}$$

where $T_{dbplant}$ is the dry-bulb temperature inside the plant. The radiation loss from the steam pipe is given by:

$$Q_{radi} = \left(\varepsilon_s \cdot \sigma \cdot T_i^4 + \rho_s \cdot G_{sky} - \alpha_{ss} \cdot G_{sky}\right) \cdot A_{di} \tag{3. 82}$$

where T_i is the pipe temperature and σ the Stefan-Boltzmann constant. Thus, the total heat loss is given as follows:

$$Q_{totali} = Q_{coni} + Q_{radi} \tag{3. 83}$$

From calculations for an uninsulated 50-mm diameter pipe with 8 bar steam pressure, it was found that 14% of the total heat loss was radiative; the rest was convective heat transfer. Cost calculations that were conducted showed that 1 m of uninsulated length of the abovementioned pipe will cost the company approximately R215 per annum for coal (see appendix D).

3.7 Refrigeration

As mentioned earlier in Chapter 3, the main refrigeration energy losses occur in the cold stores, the flow freezer, the production line and the blast freezer. At the cold stores a large percentage of the losses is caused through water vapour infiltration into the cold stores. This vapour condenses and freezes on the evaporator coils, thereby leading to inefficient operation of the evaporators. The amount of vapour ingress is a function of the cold store temperature and the outside weather conditions. Therefore, the refrigeration section is divided into weather correlations, cold stores, flow freezer, blast freezer and the refrigeration piping.

3.7.1 Weather Correlations

A year's weather data (dry-bulb and wet-bulb temperatures) at the plant was needed to obtain accurate results for the infiltration of water vapour into the cold stores over a period of one year. Weather data was obtained from the South African Weather Services for the Nortier weather station just outside Lambert's Bay. However, the data from the Nortier weather station did not correspond with the measured values at the plant in Lambert's Bay, because the plant is situated next to the sea and the Nortier weather station a few kilometres inland. Correlations between the Nortier weather station and the measured values at the plant for corresponding times were obtained and approximate correlations developed in order to estimate the plant weather conditions for one year.

Temperature and humidity measurements were taken at the plant on the 2nd and 28th of May 2002 and on the 9th and 10th of June 2002 at different hours during the day (see appendix B). The plant measurements and Nortier weather station data were plotted against time and a fifth-order polynomial curve was fitted. Adjustment formulas for temperature and humidity data were established to convert Nortier weather station data to Lambert's Bay's plant data.

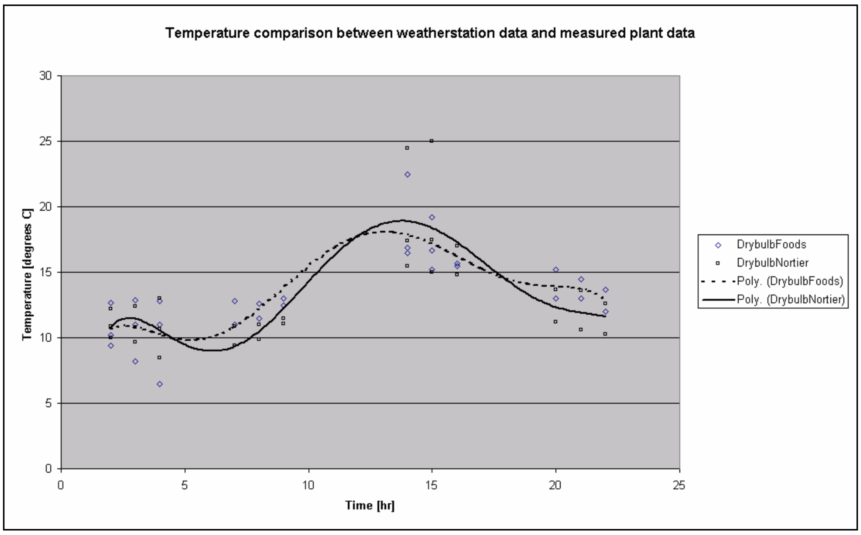

Figure 31: Dry-bulb temperature comparison

In figure 31, the measured dry-bulb temperature values at the plant (DrybulbFoods) and data from the Nortier weather station (DrybulbNortier) are indicated by fitted curves.

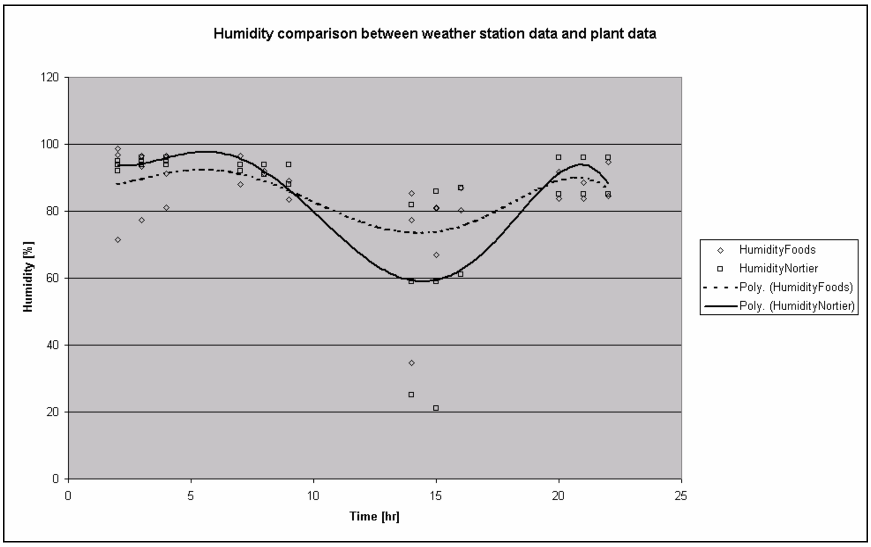

Figure 32: Humidity comparison

The relative humidity measurements at the plant (indicated as HumidityFoods) and measurements from the Nortier weather station (indicated as HumidityNortier) can be seen in figure 32. From the size of the areas between the curves it is clear that the humidity reading at the plant is higher due to its location at the harbour.

For accurate vapour infiltration calculations, an adjustment equation for the plant was used and applied to the Nortier data. This equation is a correlation of the difference between measured plant data and the Nortier data at the same time (see figure 33).

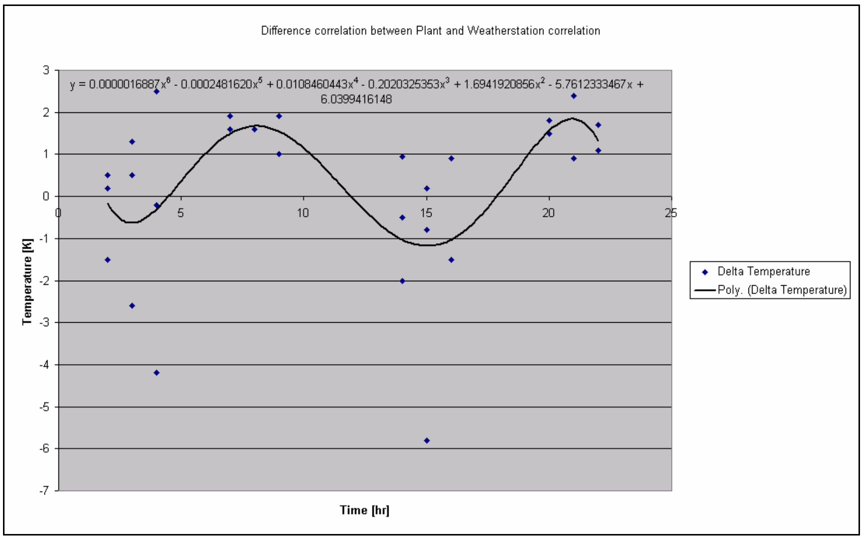

Figure 33: Temperature difference correlation

In figure 33, from the areas included between the time axis and the curve, it can be verified that the average dry-bulb temperature per day is warmer at the plant than at the weather station. For an estimate of an average temperature during the year, this correlation was adjusted to the Nortier data to find an average temperature for the plant during the year.

$$\begin{aligned}\Delta T_{plant} &= (1.6887 \cdot 10^{-6} \cdot t^6 - 2.48162 \cdot t^5 + 0.0108460443 \cdot t^4 \\ &- 0.2020325353 \cdot t^3 + 1.6941920856 \cdot t^2 - 5.7612333467 \cdot t \\ &+ 6.0399416148) K\end{aligned} \tag{3. 84}$$

with $T_{plant} = T_{Nortier} + \Delta T_{plant}$ (3. 85)

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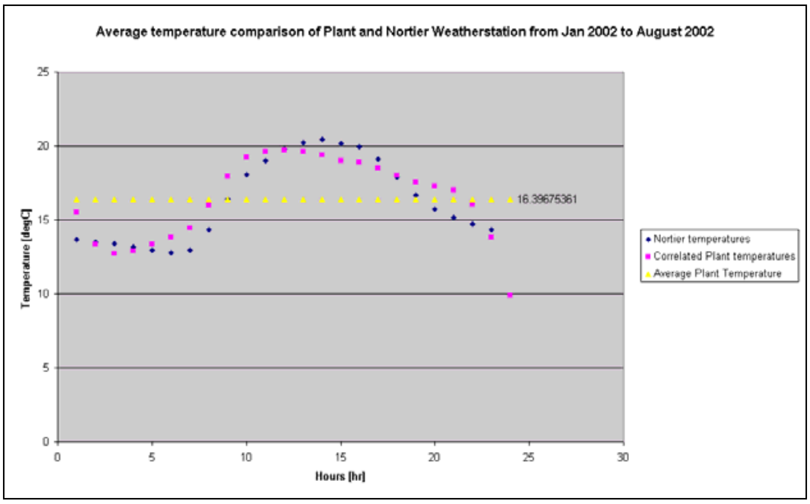

Figure 34: Average correlated temperature for plant

The Nortier data (see appendix B) was adjusted with equations (3.84) and (3.85) in order to obtain the above graph with an average plant temperature of 16.397 ^{o}C (289.547 K) for the year.

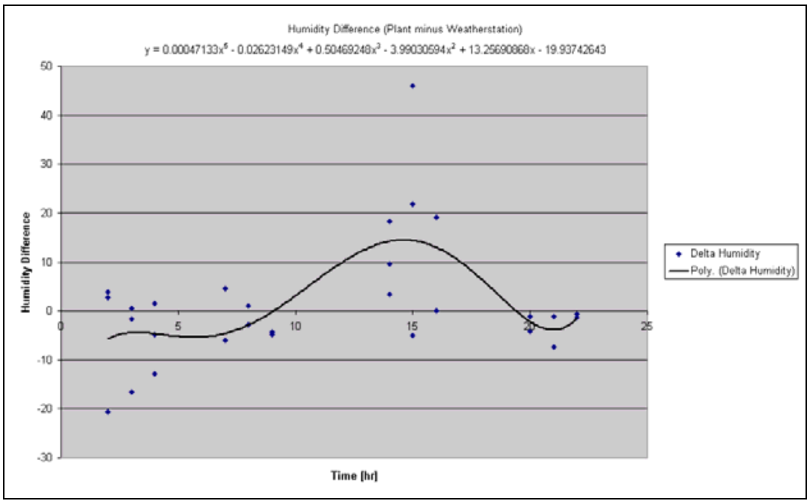

Figure 35: Humidity difference correlation

From figure 35 can be seen that the average humidity difference between the plant and Nortier weather station is positive. This correlation was used in the same manner as the temperature correlation:

$$\begin{aligned}\Delta H_{plant} &= (4.7133\cdot 10^{-4}\cdot t^{5} - 0.02623149\cdot t^{4} + 0.50469248\cdot t^{3} \\ &- 3.99030594\cdot t^{2} + 13.25690868\cdot t - 19.93742643)\%\end{aligned} \qquad (3.\ 86)$$

with $H_{plant} = H_{Nortier} + \Delta H_{plant}$ (3. 87)

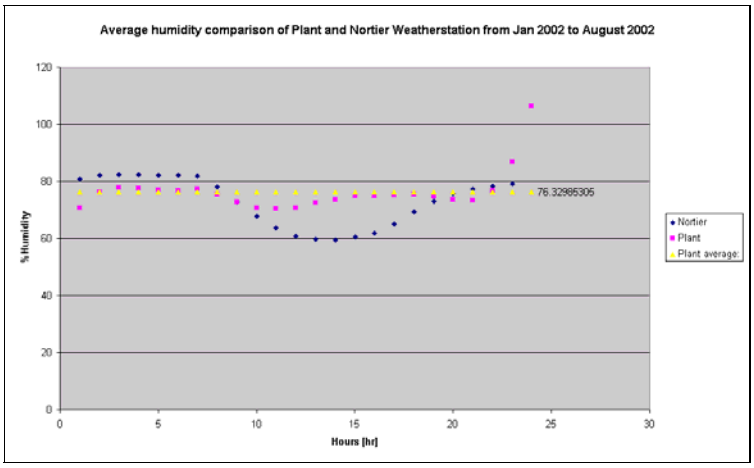

Figure 36: Average correlated humidity ratio for plant

The humidity data (see appendix B) of the Nortier weather station was adjusted with equations (3.86) and (3.87) in order to obtain the above graph with an average plant humidity of 76.330% for the year. Calculations for vapour infiltration at the cold stores were simplified by using an average dry-bulb temperature of 16.397 ^{o}C and an average relative humidity of 76.330%.

3.7.2 Cold Stores

There are three cold stores at the plant in Lambert's Bay, with a common airlock which acts as a vapour and heat-load barrier between the cold stores and the outside environment, as shown in figure 36. As can be seen, the airlock is separated from the environment by means of plastic curtains. These plastic curtains are subjected to a high rate of wear and tear and are therefore not effective in keeping moisture out of the air lock and, effectively, the cold stores.

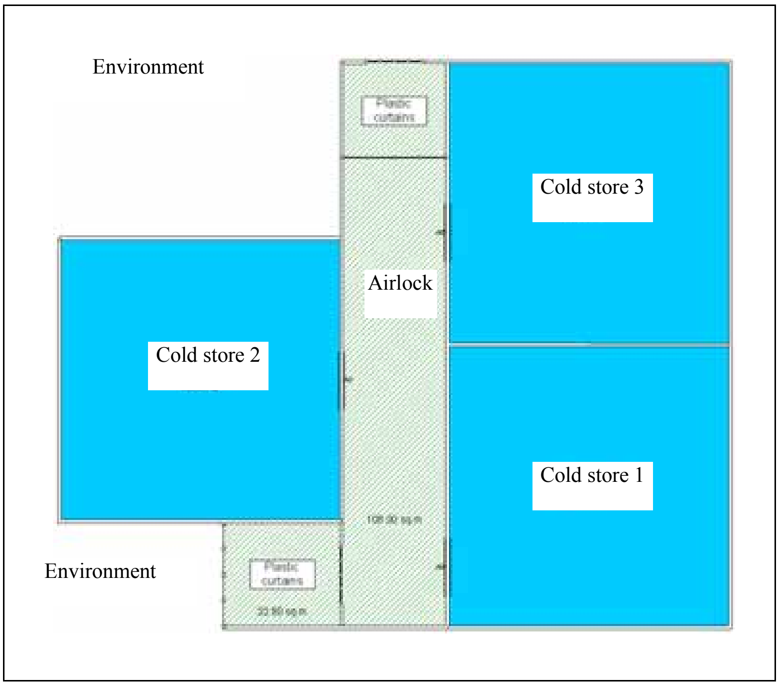

Figure 37: Cold stores and airlock

Losses at the cold stores can be ascribed to three factors:

1. Sensible heat load from the environment.
2. Vapour transfer due to infiltration and convection mass transfer from outside.
3. Sensible heat load from forklifts, forklift drivers and the product.

The first two activities will be discussed in this chapter.

Sensible Heat Load from Environment

Sensible heat enters the cold stores from the environment through radiation and natural or forced convection. Insulation panels limit the amount of heat entering the cold store. The insulation panels are constructed from chromadeck steel plates with polystyrene foam inbetween the plates. The steel plates also act as a vapour barrier. The moisture in the air wants to diffuse to the cold store due to a difference in vapour pressure and, when the outside steel plates are damaged, vapour enters through the hole and condenses at the dew-point temperature mark inside the polystyrene layer and freezes at 0 °C inside the insulating panels. The ice that builds up inside the insulating panel, fills the air volumes in the polystyrene, thereby decreasing the insulation effect of the panel and resulting into a higher sensible heat load. By calculating the new overall heat transfer coefficient and comparing it with the design value of the insulating panel, it is possible to estimate the extra amount of heat that needs to be removed.

Radiation and convection heat transfer cause a sensible heat load for the cooling room. Where the cold store's panels are subject to sunrays, radiation will play a much bigger role than on the inside where the only radiation is diffuse radiation to the surrounding air.

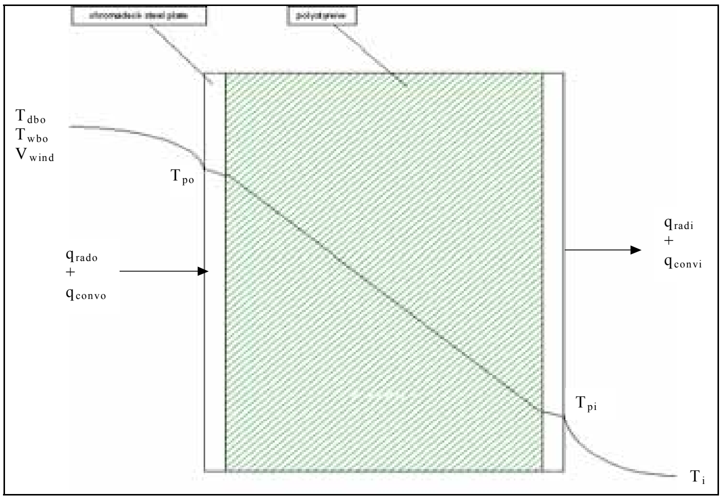

Figure 38: Insulating panel with indication of temperature gradient

By measuring the dry and wet-bulb temperatures of the outside environment, as well as the dry-bulb temperature inside the cold store and the surface temperatures of the plates, it is possible to calculate the approximate heat transfer rate, q_{total}, through the insulating panel. From this results, it is possible to calculate the overall heat transfer coefficient, $U_{overall}$.

$$q_{total} = U_{overall} \cdot (T_{po} - T_{pi}) \qquad (3.\ 88)$$

T_{po} is the outside surface-plate temperature and T_{pi} the inside surface-plate temperature.
From a correlation in Mills (1995), the emissivity of the sky can be calculated:

$$\varepsilon_{sky} = 0.55 + 1.8 \cdot \left(\frac{P_v}{P_{atm}} \right) \qquad (3.\ 89)$$

The thermo-physical properties of moist air were calculated from correlations in Kröger (1998), with the specific humidity, ω, read from the psychometric chart. The Prandtl number is calculated as follows:

$$\mathrm{Pr} = \frac{c_{pavo} \cdot \mu_{avo}}{k_{avo}} \tag{3. 90}$$

where c_{pavo} is the specific heat of the moist air, μ_{avo} the dynamic viscosity of moisture and k_{avo} the thermal conductivity of moist air. On a windless day, natural convection will take place and laminar and turbulent flow on a vertical plate equations will be used.

The Prandtl number function,

$$\Psi = \left[1 + \left(\frac{0.492}{\mathrm{Pr}}\right)^{\frac{9}{16}}\right]^{\frac{-16}{9}} \tag{3. 91}$$

is needed to determine the Nusselt number for natural convection. In natural convection, flow is initiated through density variation (buoyancy) caused by temperature difference, which is brought into account with the volumetric expansion coefficient,

$$\beta = \frac{-\left(\rho_{avo}\left(T_{po}\right) - \rho_{avo}\left(T_{dbo}\right)\right)}{\rho_{avo}\left(T_{po}\right) \cdot \left(T_{po} - T_{dbo}\right)} \tag{3. 92}$$

The Grashof number at a position x from the bottom of the wall is:

$$Gr_x(x) = \frac{\beta \cdot \left(T_{po} - T_{dbo}\right) \cdot g \cdot x^3}{\left(\nu_{avo}\left(T_{dbo}\right)\right)^2} \tag{3. 93}$$

where $\nu_{avo}(T_{dbo})$ is the kinematic viscosity of moist air calculated at dry-bulb temperature. With the Grashof number known at the full length of the plate, it is possible to calculate the Rayleigh number:

$$Ra_L\left(T_{dbo}\right) = Gr_L \cdot \mathrm{Pr}\left(T_{dbo}\right) \tag{3. 94}$$

With the Rayleigh number known, the laminar and turbulent Nusselt numbers can be calculated respectively:

$$Nu_{Lnatlam} = 0.68 + 0.670 \cdot \left(Ra_L \left(T_{dbo} \right) \cdot \Psi\left(T_{dbo} \right) \right)^{\frac{1}{4}} \qquad (3.95)$$

for $Ra_L < 10^9$

$$Nu_{Lnatlam} = 0.68 + 0.670 \cdot \left(Ra_L \left(T_{dbo} \right) \cdot \Psi\left(T_{dbo} \right) \right)^{\frac{1}{4}} \cdot \left(1 + 1.6 \cdot 10^{-8} \cdot Ra_L \left(T_{dbo} \right) \cdot \Psi\left(T_{dbo} \right) \right)^{\frac{1}{12}}$$

for $10^9 < Ra_L < 10^{12}$ (3. 96)

The convection heat transfer coefficient for convection on the outside of the cold stores is then given by:

$$h_{convo} = \frac{k_{avo}\left(T_{dbo} \right) \cdot Nu_{Lnatural}}{L_{wall}} \qquad (3.97)$$

where $Nu_{Lnatural}$ is the Nusselt number for natural flow and L_{wall} is the height of the wall. With the convection heat transfer coefficient known, the convection heat load can be calculated as follows:

$$q_{convo} = h_{convo} \cdot \left(T_{dbo} - T_{po} \right) \qquad (3.98)$$

The radiation heat transfer comprises solar irradiation, sky irradiation and surface emission. The solar irradiation, G_s, is a function of the time of day and year, but an average solar irradiation of 19 W/m^2 was used in the calculations (appendix D). The total radiation heat flux is given by the following equation:

$$q_{rado} = \alpha_p \cdot \varepsilon_{sky} \cdot \sigma \cdot T_{dbo}^{\ 4} + \alpha_p \cdot G_s - \varepsilon_p \cdot \sigma \cdot T_p^{\ 4} \qquad (3.99)$$

if T_p is higher than T_{dbo}, no surface emission will take place as heat can only travel from a high-temperature region to a low-temperature region. The Stefan-Boltzmann constant, σ, is *5.67 x 10^{-8} W/mK*, and the solar irradiation, G_s, can be obtained from solar tables. With an estimate of the total heat transfer, it is possible to calculate the overall heat transfer coefficient, $U_{overall}$:

$$U_{overall} = \frac{q_{convo} + q_{rado}}{T_{dbo} - T_{po}} \tag{3. 100}$$

This value can be compared to the design value of the insulation panels, which is obtainable from the manufacturer or can be calculated if the physical composition is known. The design overall heat transfer coefficient is calculated as follows:

$$U_{design} = \left[\frac{t_p}{k_p} + \frac{2 \cdot t_s}{k_s} \right]^{-1} \tag{3. 101}$$

where t_p is the thickness of the polystyrene – 150 mm, t_s the thickness of the steel plates, k_p the thermal conductivity of polystyrene foam as obtained from the manufacturer and k_s the thermal conductivity of steel. For the southern side of the cold stores, an extra layer of 50 mm insulation panels were added, and the design value was calculated as follows:

$$U_{designsouth} = \left[\frac{t_{pextra}}{k_p} + \frac{4 \cdot t_s}{k_s} \right]^{-1} \tag{3. 102}$$

where t_{pextra} = 200 mm. The percentage decrease, which is an indication of the decrease in sensible heat load, is calculated as follows:

$$\%Decrease = \frac{U_{design}}{U_{actual}} \cdot 100 \tag{3. 103}$$

From the measurements taken, a theoretical improvement of 951% is possible by replacing the insulating panels that are filled with ice.

Vapour Infiltration

Due to the significantly lower temperature of the cold stores, the vapour pressure is much lower in the cold stores than in the outside environment (ideal gas law). This difference in pressure causes the vapour molecules to travel from a high-pressure region (in this case the outside environment) to a low-pressure region (in this case the cold stores), resulting into frost build-up on the evaporator coils in the cold stores.

The driving force behind mass transfer is the partial pressure difference between the outside air and the cold stores. In order to estimate the annual energy costs of vapour infiltration, mass transfer calculations were done by approaching the problem as diffusion of one gas (water vapour) through a second, stagnant gas (dry air) with a lower concentration of water vapour. Fick's law for ideal gases with negligible pressure gradient states:

$$J_B = \rho \cdot D_v \cdot \frac{d}{dy}\left(\frac{\rho_B}{\rho}\right) \tag{3. 104}$$

where J_B is the diffusive mass flux, D_v the proportionality factor and $\frac{\rho_B}{\rho}$ the mass fraction.

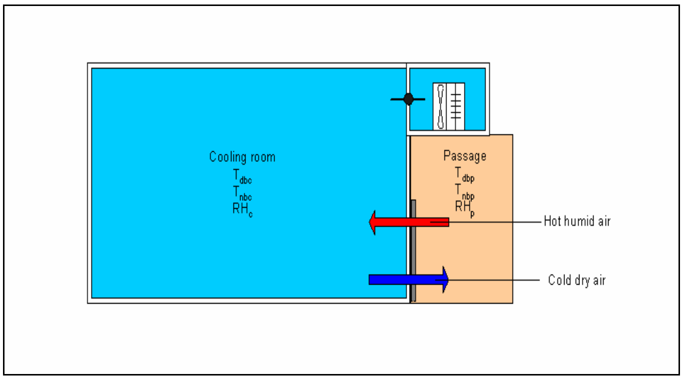

Figure 39: Infiltration effect

Let us assume the mass will be transferred from the passage to the cold store, given by the following equation:

$$J_{Bnet} = -\rho_c \cdot D_{vc} \cdot \frac{d}{dy}\left(\frac{\rho_{Bc}}{\rho_c}\right) + \rho_p \cdot D_{vp} \cdot \frac{d}{dy}\left(\frac{\rho_{Bp}}{\rho_p}\right) \tag{3. 105}$$

simplifying the problem by letting the passage and cold store each diffuse separately into a 100% dry air area, subtracting the cold store's mass flux from the passage to give net water vapour mass flow into the cold store. The interface where mass transfer occurs is the door opening where, after it is opened, the gradient tends to infinity. The mass fraction can be written as a specific humidity fraction, which makes it easier to handle:

$$\frac{\rho_B}{\rho} = \frac{m_B}{m_t} = \frac{\omega}{\omega + 1} \tag{3. 106}$$

Thus, the net mass flux can be rewritten as follows:

$$J_{Bnet} = -\rho_c \cdot D_{vc} \cdot \frac{d}{dy}\left(\frac{\omega_c}{\omega_c + 1}\right) + \rho_p \cdot D_{vp} \cdot \frac{d}{dy}\left(\frac{\omega_p}{\omega_p + 1}\right) \tag{3. 107}$$

The mass diffusivity is given by (Kröger, 1998):

$$D_v = \left(\frac{0.926}{1000 \cdot P_{atm}}\right) \cdot \left(\frac{T_{db}^{\ 2.5}}{T_{db} + 245}\right) \tag{3. 108}$$

with P_{atm} in Pa and T_{db} in K. Finally, the mass transfer rate can be calculated:

$$m_{transfer} = A_{door} \cdot J_{Bnet} \tag{3. 109}$$

The above calculations are for diffusion only and do not include convection mass transfer. As the forklift enters the airlock through the plastic curtains, it generates air movement inside the airlock and if the doors of the cold stores are open, convection mass transfer occurs, resulting in a larger amount of water vapour inside the cooling room that needs to be removed by the evaporators.

The energy cost of 1 kg of water vapour was calculated. The energy needed to freeze water from vapour to ice is the difference between the enthalpy value of the water vapour at the outside air's dry-bulb and wet-bulb temperature and the enthalpy value of ice at the evaporator temperature.

The enthalpy of vapour and ice is given by the following equations respectively:

$$h_v = 2501\frac{kJ}{kg} + 1.805\frac{kJ}{kg \cdot K} \cdot \left(T_{db} - 273.15K\right) \tag{3. 110}$$

$$h_{ice} = c_{pice} \cdot T_{dew} \tag{3. 111}$$

where c_{pice} is 1.860 kJ/kgK and T_{dew} is the temperature of the evaporator coil, assumed to approach dew point. The energy needed to freeze a certain mass of water, m_{water}, is:

$$E_{freeze} = m_{water} \cdot (h_v - h_{ice}) \cdot t_{operating} \qquad (3.\ 112)$$

where $t_{operating}$ is the period that the cold stores are in operation during the year and m_{water} is the average mass of water condensed through the year. The number of kWh units can then be obtained by dividing E_{freeze} by 1 kWh, whereas the electricity costs are obtained by using an average cost per unit for the year and assuming the cooling rooms are operative 24 hours per day. Payback calculations were done (see appendix G) to motivate the capital expenditure for air curtains, which can be assumed to work with an effectiveness of 70% to 90%.

3.7.3 Flow Freezer

The flow freezer (see figure 40) is on the production line and freezes the French fries as they enter from the precooler, which uses outside air to precool the French fries. The fries enter at approximately 20 °C and exit at approximately –16 °C.

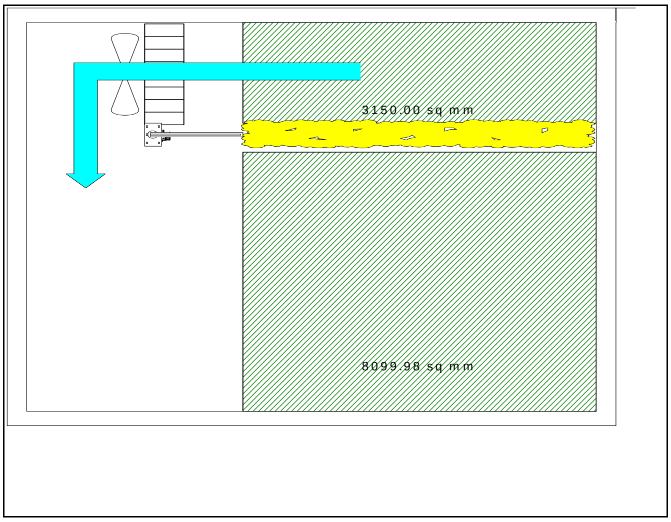

Figure 40: Schematic representation of flow freezer

Air inside the flow freezer is circulated across a plate heat exchanger by means of an axial fan. Losses that occur at the flow freezer are almost similar to those at the cold stores, except that in the flow freezer's case, the largest energy losses are due to vapour condensation on the heat exchanger, resulting in inefficient heat transfer from the heat exchanger to the circulating air. This effect causes the fan to work harder, resulting in shorter motor life and higher fan-power requirements.

To keep moisture out of the flow freezer one must keep the inside pressure slightly higher than the outside pressure and add pressure drop obstructions between the outside air and the circulating air. Moisture infiltration at the flow freezer is a significant energy loss factor.

3.7.4 Blast Freezer

The blast freezer is a batch freezer. It removes sensible heat from the packaged product as it is removed from the production line. The temperature of the product needs to be lowered from –6 ^{o}C to –25 ^{o}C before it is stored in the cold stores, especially if it is to be transported in the next two days. The blast freezer works on the same principle as the flow freezer, except that the packaged product is static inside an enclosed volume. To minimise energy losses, the insulation must be kept in good shape and the door must seal properly. Valuable refrigeration capacity is destroyed when hot, humid air infiltrates the freezer and cold, dry air leaves into the factory environment, which is also undesirable. This effect can be minimised through plastic curtains at the openings of the blast freezer.

3.7.5 Refrigeration Pipes

Heat loss calculations of the refrigeration insulation were conducted by measuring the surface temperature on various points on the insulation and assuming natural convection in the still air of the plant. The calculations that were done to determine the energy loss are shown below.

Assuming natural convection for the refrigeration pipes inside the plant, measurements of the outside temperature on the refrigeration pipes were estimated by means of a Raytek infrared thermometer and the physical dimensions of the insulation were taken. The wet and dry-bulb temperatures, $T_{dbplant}$ and $T_{nbplant}$, for the inside of the plant were measured with a wet and dry-bulb sling thermometer. The refrigeration pipes were divided into segments to simplify heat gain calculations.

For natural convection over a horizontal cylinder, the Grashof, Nusselt and Rayleigh numbers were calculated with:

$$\beta = \frac{1}{T_{dbplant}} \tag{3. 113}$$

$$k = 0.026 \frac{W}{m \cdot K} \tag{3. 114}$$

$$\Pr = 0.69 \tag{3. 115}$$

$$\nu = 14.7 \cdot 10^{-6} \frac{m^2}{s} \tag{3. 116}$$

$$g = 9.807 \frac{m}{s^2} \tag{3. 117}$$

where β is the volumetric coefficient of expansion, k the conductivity, Pr the Prandtl number and ν the kinematic viscosity coefficient for air. The Grashof, Rayleigh and Nusselt numbers are given by the following equations:

$$Gr_{Di} = \frac{\left(\beta \cdot \Delta T_i \cdot g \cdot d_i^3\right)}{\nu^2} \tag{3. 118}$$

$$Ra_{Di} = Gr_{Di} \cdot \Pr \tag{3. 119}$$

$$Nu_{Di} = 0.36 + \frac{\left(0.518 \cdot Ra_{Di}^{0.25}\right)}{\left[1 + \left(\frac{0.559}{\Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}} \tag{3. 120}$$

With the Nusselt number known, the heat transfer coefficient can be found:

$$h_{ci} = k \cdot \frac{Nu_{Di}}{d_i} \tag{3. 121}$$

and the heat gain is calculated as follows:

$$Q_i = h_{ci} \cdot A_i \cdot \Delta T_i \tag{3. 122}$$

The area, A_i, was estimated as a percentage of the full area of the segment at a certain temperature. The total heat gain for all the refrigeration pipes were added up to determine the number of kWh units lost. Assuming fairly uniform operation over 24 hours, an average price per kWh unit was estimated in order to calculate the savings possible per time period, S_p. For I_p, the interest rate for the duration of the period and n, the number of periods, the payback period can be calculated. Calculations done in appendix A illustrate that, with the current savings and a payback period of 10 years, R3 682 can be spent, which is much less than necessary.

3.8 Product Lifecycle Analysis

Improving the efficiency of the production line not only increases productivity, but also leads to energy and salary savings (Energy Research Institute, 2000). In the figure below a flow diagram of the production line is shown, indicating the various processes in the production line. Specifications for the finished product allows 2% of the finished product to be defective. It is estimated that 60% of incoming defects are removed at inspection belt 1 and 40% is removed at inspection belt 2. Currently, 10 workers (six at inspection belt 1 and four at inspection belt 2) are needed to remove the defects manually. From estimations in 2001, the incoming defects before removal for 60% of the time are 15% of the incoming product and the inspection belts can reach specification (2% defects allowed). For 15% of the time the incoming defects are less than 10% and the inspection belts meet specification, and for the next 25% of the time the incoming defects are 20% and 4% defects are in the final product, unable to reach specification.

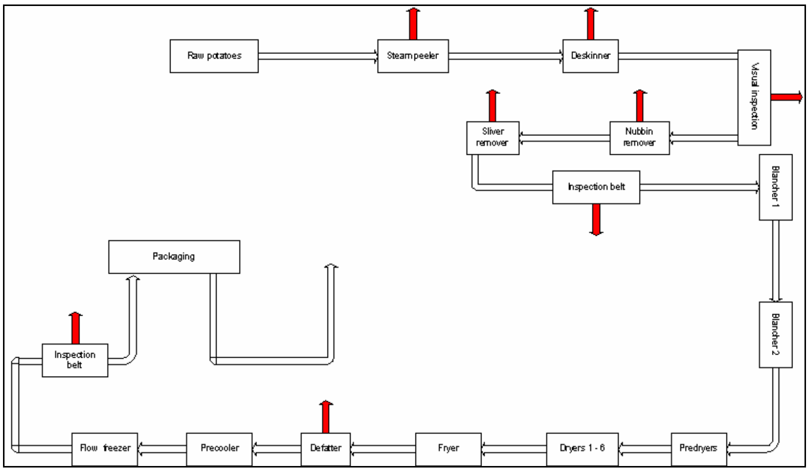

Figure 41: Production process line

The total defects removed from the production line in a year amount to 12.55%, of which 60% (7.53% defects) is removed at inspection belt 1 and 40% removed at inspection belt 2. By maximising the percentage of defects removed at inspection belt 1, less defects will move through the production line absorbing unnecessary energy. This will lead to less product flow through blancher 1, blancher 2, the predryers, dryers 1 to 6, the fryer, the de-fatter, the precooler, the flow freezer and the shaker. In turn, this will lead to less steam, refrigeration energy and electricity wasted on defective products that are thrown away. It also shortens inspection line 2, because fewer workers are needed to pick out defects which in turn leads to less exposure of frozen product to the warmer environment. It will also save the salaries of six workers on inspection belt 1 and three workers on inspection belt 2. This could be achieved by implementing an optical sorter.

Optical Sorter Modification in Production Line

An optical sorter will practically remove all the defects at inspection belt 1 (see figure below), whereas only 0.1% of the defects will need to be removed at inspection belt 2 or it could be left to fit into specifications.

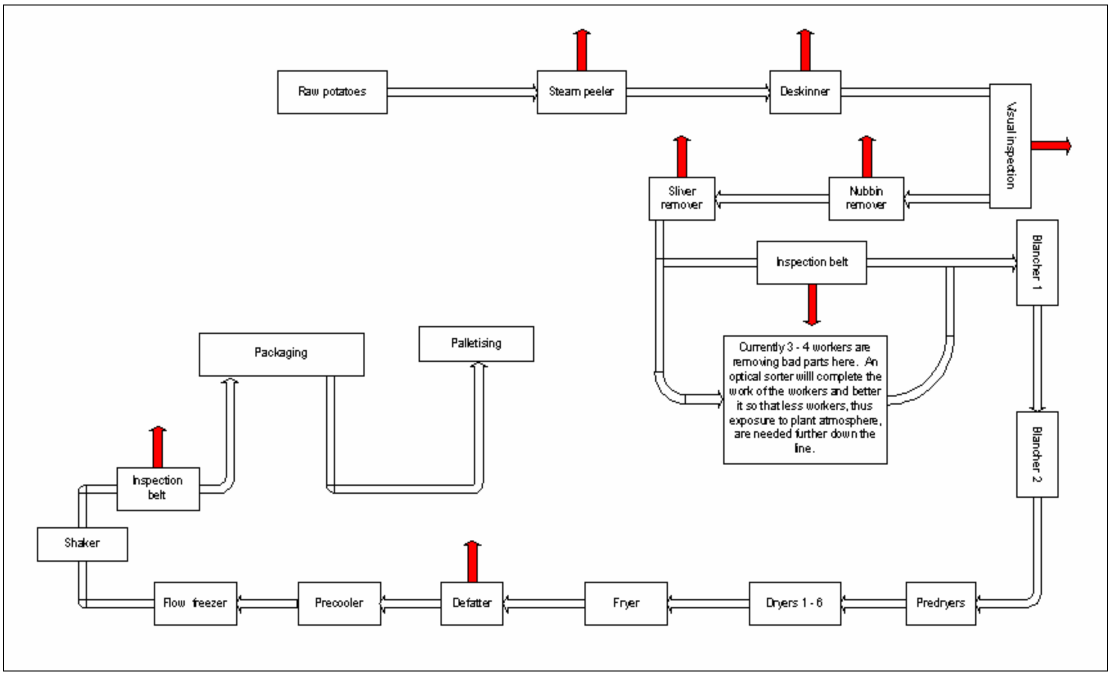

Figure 42: Production line with optical sorter modification

The optical sorter will lead to the removal of 4.92% of the defects from the production line – defects which were previously processed. The coal savings amount to R110 801.30 per year, the electricity savings amount to R5 151.46 and savings from less labour amount to R587 500 per annum (see appendix I). By estimating maintenance at 10% of the optical sorter's value for the first five years and subtracting this value from the total savings, the payback period of the equipment can be determined:

$$n = \frac{\ln\left(\dfrac{1}{1 - \dfrac{1.3 \cdot 10^6}{573452.76} \cdot 0.17}\right)}{\ln(1 + 0.17)} \qquad (3.\ 123)$$

$$n = 3.1003$$

With the inclusion of maintenance estimates, the optical sorter will be paid off in three years' time.

3.9 Waste Utilisation

Potato-related wastage caused by processes in the production line can be utilised in many different ways. One of them is the production of ethanol from potato waste. The figure below shows where the waste of the de-skinner exits the plant. Currently, outside contractors are employed to remove the potato waste.

Figure 43: Potato waste from steam peeler at plant

However, these waste products can be used for alcohol production, where the alcohol can be used to generate steam or dry a part of the potato waste and sell it as animal food. Since the oil crisis of 1973 and 1979, the significant rise in oil prices have led to research of alternative energy sources, of which the potential alcohol yield of potatoes were determined. The potential alcohol yield of 1 ton potatoes is 600 litre. The daily potato wastage at the plant is approximately 10 ton. Let us assume the alcohol conversion process can be done at 80% efficiency. The alcohol yield is 600 l/ton potatoes. The volume of alcohol that can be produced daily is as follows:

$$\begin{aligned} V_{alc} &= \eta_{con} \cdot \eta_{util} \cdot Yield_{alc} \cdot \dot{m}_{waste} \\ V_{alc} &= 4560 l / day \end{aligned} \tag{3. 124}$$

Assuming that the alcohol conversion process is an optimised one, which (according to Misselhorn, K. Oktoberagung der Wersuchs - und Lehranstalt Fur Brauerei) needs an energy input of 8 MJ per litre of dry ethanol, where the higher heating value of ethanol is 30.61

MJ/kg of ethanol, there is a positive excess amount of energy available for other use. Let us assume that after distillation, the ethanol is at a temperature of 20 ^{o}C, with a density of 789 kg/m^3, then the energy potential of the ethanol amounts to:

$$
\begin{aligned}
Energy_{produced} &= E_{out} - E_{in} \\
Energy_{produced} &= V_{alc} \cdot \rho_{alc} \cdot \left(HHV_{alc} - \frac{e_{in}}{\rho_{alc}} \right) \\
Energy_{produced} &= 0.05278 \frac{L}{s} \cdot 789 \frac{kg}{m^3} \cdot \frac{0.001 m^3}{L} \cdot \left(30.61 \frac{MJ}{kg} - 8 \frac{MJ}{L} \cdot 789 \frac{kg}{m^3} \cdot \frac{0.001 m^3}{L} \right) \\
Energy_{produced} &= 1.011809 MW
\end{aligned}
$$

(3. 125)

The amount of coal that can be saved annually through the production of ethanol amounts to R380 000.

4 CONCLUSION

Analyses of the steam dryers in section 3.6.1 identified the source of ineffective operation and it was found that improved housekeeping could lead to an increased effectiveness to accommodate a higher production rate.

The implementation of a flash-steam heat recovery system at the boiler will lead to a monthly energy saving of almost R14 000,00 with a payback period of 3.3 years at a capital cost of R44 206,00.

The recovery of waste steam at the steam peeler with the implementation of a finned-tube heat exchanger in the deceleration duct will lead to a monthly energy cost saving of R8 675 with a payback period of 14.19 months at a capital cost of R110 836,00.

Measurements and heat load calculations for the insulating panels of the cold stores indicated that damaged insulation panels have a dramatically lower insulation effect due to ice formation inside and, thus, a higher heat conduction coefficient. Vapour infiltration also leads to a higher heat load and maintenance cost (forklift accidents on slippery floor). The implementation of air curtains will decrease vapour infiltration by 80% and have a payback period of 1.791 years, based only on energy savings at a capital cost of R9 900,00.

Analyses of production line modifications (see section 3.8) showed that, through the improvement of the inspection process earlier in the line by implementing an optical sorter, labour and energy savings will amount to R573 452 per annum and have a simple payback period of 3.1 years.

A preliminary study of waste utilisation showed that an in-depth analysis of ethanol production from potato waste is needed to prove the feasibility.

Future research can be conducted on the acquisition of an optical sorter and on improvements on the production line to minimise energy losses due to radiation.

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A APPENDIX: ELECTRICITY, COAL AND PRODUCTION RECORDS

A.1 Electricity Readings

Date

23/02/2002

Equipment	Current [Amp]	Watt
Transformer 1	1030	391400
Transformer 2	750	285000
	1780	676400
FES Compressor	280	106400
M&M Compressor 2	320	121600
M&M Compressor 3	340	129200
M&M Compressor 4	147	55860
Boiler electricity	45	17100
Refrigeration electric motors	115	43700
Production line	533	202540
Total electricity		676400
Total refrigeration		456760
Coal energy usage rate	[kg/day]	Watt
Calorific value of coal [J/kg]	27320000	
Coal usage: 13 bags at 850kg each	11050	3494050.926

Table A. 1: Electricity readings and coal usage

A.2 Electricity and Coal Consumption

Electricity usage and costs		
Off-peak	0.0859	Energy unit
Peak	0.3923	J
Standard	0.148	3600000

	Usage				Coal price	
	Off-peak	Peak	Standard	Electricity cost	R/kg	Calorific value
December	346863	126359	315192	126014.5834	447.5	J/kg
January	408974	142368	380440	147286.953		28500000
February	361300	143830	354365	139906.199		
March	352922	134165	337495	132898.1893		
April	442585	158377	418115	162030.3686		
May	479331	172953	422610	171570.2748		
Last 4 months	1636138	609325	1532585	606405.0317		
			Average	151601.2579		
			Unit price	0.160507498		

	Energy comparison							
	Coal usage	Coal cost	Elec. MJ	Elec. cost/MJ	Coal MJ	Coal cost/MJ	Unit elec.	Unit coal
January	1420799.5	635807.7763	3354415.2	0.043908385	40492785.75	0.015701754	0.86685943	0.91002922
February	1085001.5	485538.1713	3094182	0.045215892	30922542.75	0.015701754	0.799609078	0.694948913
March	1133679	507321.3525	2968495.2	0.044769548	32309851.5	0.015701754	0.767128666	0.726127097
April	1469148	657443.73	3668677.2	0.044165883	41870718	0.015701754	0.948072089	0.940996677
May	1561268	698667.43	3869618.4	0.044337776	44496138	0.015701754	1	1
		2348970.684	13600972.8	0.044585416	149599250.3	0.015701754		
	Average	587242.6709	3400243.2	0.044585416	37399812.56	0.015701754		

Table A. 2: Electricity and coal consumption data for January 2002 till May 2002

A.3 Electricity, Coal and Production Comparison

Electricity, coal and production comparison				
Month	Electricity usage MJ	Coal usage MJ	Production rate raw kg/month	Yield
January	3354415.2	40492785.75	2543684	55.9%
February	3094182	30922542.75	1980374	58.1%
March	2968495.2	32309851.5	2042707	54.5%
April	3668677.2	41870718	2609655	56.0%
May	3869618.4	44496138	2719978	57.4%
Average	*3391077.6*	*38018407.2*	*2379279.6*	

Electricity, coal and production comparison on a unit basis			
Month	Electricity	Coal	Production
January	0.989188569	1.065083698	1.069098394
February	0.912448008	0.813357135	0.832341857
March	0.875384037	0.849847584	0.858540123
April	1.081861766	1.101327517	1.096825695
May	1.14111762	1.170384066	1.143193931

Electricity, coal and production comparison on a unit basis				
Month	Dry-bulb [^{o}C]	Humidity	Unit Dry-bulb	Unit Humidity
January	18.5375	75.625	1.015753425	1.051564311
February	21.0125	64.29166667	1.151369863	0.893974508
March	19.70833333	69.75	1.079908676	0.969872538
April	17.275	73.25	0.946575342	1.018539977
May	14.71666667	76.66666667	0.806392694	1.066048667
Average	*18.25*	*71.91666667*		

Table A. 3: Electricity, coal and production for January 2002 till May 2002

A.4 Walkthrough Audit Checklist

Walkthrough audit checklist

Auditor: _________________

Date: _________________

Fuel, gas or oil leaks	Steam leaks	Condensate leaks	Water leaks	Damaged or leaking insulation	Excess utility usage	Equipmant running and not needed	Burners out of adjustment	Leaks of or excess of HVAC	Location

B APPENDIX: NORTIER TEMPERATURE AND HUMIDITY DATA

B.1 Nortier Temperature Data

Hours of the day – Nortier temperature data January 2002 – August 2002

Day	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
1	14.7	14.7	14.7	15.1	13.7	13.5	16	18.8	19.8	21.4	21.8	21.5	22	21.8	21.6	21.7	20.5	19.5	19.1	17.6	17.4	16.7	16.3	15.7
2	14.7	14.6	13	13.1	12.6	12	16.1	18.4	19.6	21	23.8	23.9	23.2	24.1	24.3	25.3	24	25.2	22.7	21.2	20.3	19.6	19.5	16.7
3	14.9	13.8	13.7	13.8	14.1	13.6	14.9	16.7	17.8	20	19.1	23.4	22.4	23.6	23.9	26.3	22.1	21.9	21.4	19.2	17.6	17	16.5	14.1
4	13.1	12.5	13.2	14.1	13.8	11.7	14.6	17.7	21.8	20.6	21.5	21.6	22.3	23.2	23.9	25.2	21.6	20.5	18.3	17.9	17.2	16.9	16.1	15.4
5	15.3	13.8	14.9	15.7	15.7	15.7	16.2	16.6	17.4	19	21	22.1	22.8	23	23.1	22.1	22.5	21.5	20.2	18.3	17.6	17.5	16.5	17.6
6	17.8	17.8	16.4	16.8	16.5	15.4	14.5	14.9	16.6	16.8	17.2	18.7	18.6	19.9	20.1	20.2	20.1	19.6	18.1	16.5	15.4	15	12.9	11.8
7	11.5	11.7	13.4	13.4	13.1	13.2	12.8	14.6	16.3	17.4	18.2	18.5	18.2	19.6	19.5	19.3	19.3	18.6	17.1	15.8	15.1	14.3	13.7	13
8	13	12.4	11.5	10.9	11.1	11.4	14.1	17.3	19.4	20.7	20.4	21.7	21.1	22.3	22.3	22.1	20.7	20.5	20.7	18.8	17.5	16.5	15.9	14.9
9	14	13.7	13.1	11.2	10.8	10	13.5	16.4	19.6	19.8	20.3	20	20.7	21.8	21.5	21.6	20.9	19.9	17.9	16.5	15.7	15.1	14.8	14.6
10	15.2	15.5	14.6	15.3	15.6	15.4	15.9	17.5	19.4	21.3	22.2	21.7	21.7	21.8	21.6	21.9	22	21.2	19.8	18.1	17.3	16.2	14.9	13.8
11	15.3	15.1	14.3	16.1	16.9	17.1	17.9	19.1	19.9	17.5	17.4	17.1	17.9	20.2	19.4	19.6	18.1	17.2	17.9	16.3	15.9	15.3	15.6	15
12	15	14.6	14	14.1	13.7	12.6	13.8	16.1	17.8	19.4	20.4	20.6	20.7	21.4	20.9	21.1	20.8	19.5	18.5	17	15.6	15	14.5	13.9
13	13.9	13	12.8	12.9	11.8	11.5	14.3	18.2	21.3	20.5	21.8	22	23.2	22.5	22.5	22.2	21.8	20.5	19.7	18.1	17.1	16.4	15.5	15
14	14.1	13.5	11.9	11	10.6	10.4	12	16.6	19.6	20.5	21.3	21.4	21.2	21.8	21.6	21.4	21	20.2	18.9	17.5	17.2	16.7	16.2	15.8
15	15.1	14.5	13.2	13	12.2	14.2	14.2	15	18.1	19.8	20.4	20.6	20.9	22.3	22.9	22	21.3	20.8	19.3	17.9	17.2	16.8	16.5	15.7
16	15.3	15.6	13.1	13	13.8	15.2	16.5	17	16.7	18.9	20.7	21.9	22.6	22.5	22.1	21.9	21.1	20.6	19.5	18	17.4	18	16.3	16
17	15.5	16.1	15.9	15.1	14.4	14.4	15.1	16.4	17.7	18.9	18.3	16.2	17.8	17.6	16.8	16.5	17.8	18.1	17.3	16.8	15.6	15.1	14.8	15.3
18	15.6	15.3	14.7	14.5	13	12.9	14.5	17.6	19	19.8	20.3	20.3	20.8	20.9	21	20.7	20.6	19.8	19.3	18.3	18.1	18	17.4	16.9
19	16.4	15.6	15.4	14.6	14.2	13.1	15.4	17.5	19.8	21.7	22.3	23	22.3	23.7	23.1	23.3	22.1	21.1	19.2	18.4	18	17.5	16.6	16.1
20	16.2	16.2	15.1	14.4	12.5	12.1	15.1	17.5	19.3	20.3	20.7	21.8	21.9	22.8	22.5	22.2	21.5	20.9	19.8	18.3	18.1	18.1	18.1	17.7
21	16.6	16.1	15.7	15.4	15.5	15.5	16.7	19.7	21.7	21.7	22.8	23.4	23.4	22.8	23.3	22.2	21.7	21.5	20	19	19.1	18.5	18.2	18
22	17.5	16.7	16.5	16.2	16.2	16.8	17.4	18.3	20.9	21.8	23.2	23.9	24.5	23.9	24.4	22.4	21.8	21.2	19.8	19.3	19	19.5	19.6	19.2
23	18.2	17.7	17.9	17	16.5	16.2	17.9	21.5	22	21.6	23.4	24	23.9	23.5	23.3	23.1	22.2	22.7	21.4	19.4	18.7	17.2	16.6	15.8
24	15.1	14.8	14.8	14.1	12.8	13.9	14.9	17.7	22.2	20.9	21.5	22.5	23.5	24.5	25.2	26.1	25.1	23.9	22.9	21.6	20.7	18.9	18.2	17.9
25	17.5	17.2	16.9	16.1	15.5	15.9	17.2	19.3	19.8	17.4	18.8	20.2	20.7	20.7	21.9	21.4	22.8	21.6	18.7	17.6	17.5	17.5	16.8	16.8
26	17.2	17.5	17	16.5	16.5	16.5	16.4	17.3	19.1	21.2	22.3	22.7	23	22.5	22.4	22.5	21.9	21.4	20.2	19.3	18.9	18.6	17.8	17.4
27	17.5	18.3	18.6	18.8	18.7	18.5	18.6	20	21	22.3	22.2	22.2	21.9	21.6	21.1	19.6	19.6	19.7	19.7	19.3	19.3	18.1	17.5	17.2
28	17.3	17.6	18.4	16.9	15.7	15.6	17.5	20.8	21.6	23.9	23.4	23.8	23.2	22.6	22.3	22.9	21.6	21.6	20.5	19.3	18.8	18.6	17.6	15.8
29	17.6	17.9	18.1	17.6	15.7	16.6	18.1	19.5	21.5	22.1	22.5	22.8	23	23.2	23.9	24.1	23	22.4	21.1	20.3	19.5	19	18.1	18.8
30	18.5	18.8	19	18.9	19	19	19.1	19.6	21.9	21.7	23.8	24.6	24	24.5	24.5	25.1	23.8	23.3	22.4	21.4	19.8	19	18.5	17.6
31	16.8	16	15.3	14.5	14.7	15	16.9	20.9	25.4	24.7	24.9	25.9	26.1	27.3	27.9	25.6	26.2	25.4	25.6	23.2	22.8	23.1	22.1	20.8
1	23.1	22.9	22.2	23.5	22.8	23.9	26.7	28.7	29.4	31.4	33.9	34.9	31.7	32.7	33.7	30.9	29.3	30.4	28.4	26.1	24.5	24.8	23.9	23.2
2	22.9	21.9	21.4	21.1	20.7	19.9	21.5	23.4	26.2	26	26.8	26.5	26.6	27.7	26.5	24.9	25.2	25.3	24.5	20.1	18.9	18.2	17.2	16.1
3	15.8	15.2	15.9	16.3	16.5	16.6	16.5	17.4	19.8	21	21.1	22.8	22.8	22.2	23.9	23.5	24.3	22.5	21.5	19.7	18.7	18.2	17.7	16.9
4	16.6	16.2	15.9	15.3	14.3	12.9	15.1	16.6	18	20	21.8	22.1	23	23.3	23	22	22.6	22.1	20.7	18.8	18.5	17.7	17.9	17.5
5	17	17	16	14.9	15.1	14.9	15.4	18.5	20	20	22.3	21.8	22.5	22.5	21.5	22.1	20.8	19.5	20	16.5	16.4	16.9	16.8	17.6
6	15.7	15.3	15.3	15.6	15.8	15.4	16.1	17.4	18.1	15.2	15.1	17.6	21.8	20.1	20.6	19.5	18.5	17.6	17	15.6	14.7	14.8	15.2	15.3
7	13.9	13	13.1	11.2	10	10.1	11.3	17.1	20.5	24.4	22.8	22.9	23.3	23	21.9	23.2	22.4	20.7	20.2	18.3	18	17.2	17.9	17.6
8	18.6	18.2	19	19	18.6	18.2	20.1	25.8	27.8	25.8	27.7	28.5	29.5	27.5	27.4	29	28.3	27	25.8	22	21.9	20.9	20.5	20
9	18.9	21.4	21.4	20	15.7	13.9	17.7	23.9	27.6	29.8	28.8	30.1	28.7	25.4	22.5	22.7	22.1	19.5	16.2	15.5	15.3	16.2	16.1	15.9
10	16	16.4	16.5	16.4	16.3	16.4	16.5	17.1	17.5	18	19.5	20.8	21.9	22.4	21.8	21.4	19.9	19.2	18.8	18	16.9	16.9	16.9	16.6
11	16.1	15.9	15.3	15.5	15.7	16.2	16.1	16.7	16.5	16.9	19.4	21.1	20.8	20.6	22.4	21.7	21.5	20.2	19.5	18.3	16.9	16.6	15.2	14.4
12	14.1	13.3	12.9	12	11.7	11.5	12	14.9	17.1	17.6	19.8	20.2	20.6	20.6	20.6	19.8	18.8	18	16.1	16.2	15.5	15	14.6	13.8
13	13.5	12.4	12.1	11.9	11.5	11.3	10.7	14.2	16.6	18.3	19.7	19.7	18.7	20.1	20.3	20.6	20.1	19.1	17.3	16.2	16.1	15.4	15.1	13.4
14	13.4	12.4	12.2	13.1	11.3	10.8	12.2	15.3	17.8	19.4	20.7	21	21.7	21.1	21.6	21.7	21.2	20	18.8	17	16.5	15.7	15.3	13.7
15	13.4	14.2	14.9	15.3	16.3	16.8	17.1	17.8	19	18.8	17.1	17.8	19.3	21.2	20.8	21.2	20.8	19.5	18.1	16.3	15.7	15.4	14.9	14.4
16	13.7	12.6	12.2	11.8	10.9	10.4	11.4	16.5	19.8	20.1	20.2	21.3	22.9	22.8	22.7	22.5	23	22.1	20.4	18.5	18	16.6	15.6	15.1
17	15.3	14.1	15.3	15	14.4	14.8	17	23.5	25.7	28.9	24.6	27.5	27.2	27.6	25.5	27.7	27.7	28.3	26.2	24	22.2	21.5	23.7	22.7
18	21.9	21.5	20	18.4	21.9	19.4	22.4	26.5	30.6	33.3	35.3	36.9	38.2	30	28.9	33	32.8	27.5	20.7	17.5	16.9	17.1	16.9	15.8
19	16.4	15.4	15.2	16.1	18.4	21.2	22.7	25.4	30.8	28.8	25.1	24	26.2	26.6	25	24.7	25.1	25.9	26.6	24.5	20.3	19.2	18.9	19.1
20	20.7	18.2	19.6	19.2	18.5	16.3	18.9	22.2	21.6	27.8	22	24	33	33.8	26.8	29.4	29.3	28.8	28.2	25.1	24.4	24.8	25	24.3
21	21.9	20.3	18.9	19.2	18.4	18.2	18.4	20.2	20.2	21.9	26.1	27.7	28.5	28.4	27.6	28.8	27.8	28	23.8	22.7	23.4	21.1	19.1	18.4
22	17.9	17.5	17.7	17.3	17.4	17.5	16.8	16.6	16.4	17.6	19.1	20.7	22.7	23.2	22.6	21.6	20.4	18.6	18.4	18.3	18.4	18.6	18.7	18.6
23	18.6	18.2	17	16.2	15.9	16	15.8	17.5	20.9	21.3	20.5	21.6	20.4	24.3	22	21.6	19.9	18.7	18.5	20.5	21.1	20.9	23.6	24.1
24	20.9	21.9	21.2	19.8	18.7	18.3	18.1	23.9	25.5	29.5	30.7	29.3	28.7	30.2	29.1	29.9	28.9	27	25.9	25.9	23.3	22.3	22.5	21.8

Hours of the day – Nortier temperature data January 2002 – August 2002

Day	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
25	20.9	19.9	19.4	18.9	19.8	21.2	21	23.2	26.9	29.9	28.2	30.7	30.4	29.7	33	31.8	29.9	29.2	27.8	26.5	26.3	25.5	24.8	25.4
26	23.5	22.2	22.4	22.6	23.9	26.5	25.8	29.7	33.7	35.3	36.4	38.3	40.3	40.3	33.7	35.3	29.1	29.7	28.6	28.6	27.8	27.8	25.1	23.4
27	20.9	20.4	20.5	21.3	21.2	21.3	21.4	22.9	23.8	28.7	25.2	28.2	24.2	27.6	22.5	23.1	25.5	26.2	26.6	22	22.2	22.5	22.3	22.3
28	23.3	20.7	20.3	20.4	21.5	21.8	20	21.9	22	18.9	20.2	20.9	24.4	29.2	26.2	25.6	24.6	25.2	21.2	20.5	20.1	19.4	18.8	18.7
1	18.5	18.1	18.2	17.6	18.3	18.7	18.5	20.3	21.3	23	24.3	24.7	24.4	24.4	24.5	23.9	24	22.6	21.2	20	19.6	19.4	19.1	18.9
2	18.6	18.4	17.9	18	17.4	16.5	16.3	17.1	17.4	18.1	20.9	22.1	23.2	23.2	23.1	23.1	22.8	21.9	20.4	19.4	19.6	19.3	19	18.9
3	16.3	15.8	16.1	16.1	16.6	16.1	15.9	18.1	21.8	21.8	23.4	24.7	24.9	24.7	24.5	24.4	24.5	23.8	21.9	20.2	19.4	18.2	17.3	16.4
4	16.4	15.9	15.8	15.6	15.3	15.6	16.4	18.8	23.3	25.5	25.7	27	27.3	22.9	23.3	21.5	20.5	20.4	21.7	19.5	18.6	18.4	16.9	17.9
5	17.1	16.6	16.1	16.5	16.6	16.7	17.4	22.7	23.5	22.1	20.8	22.9	22.5	24.8	22.9	21	20.5	20.7	19.6	19	18.2	17.7	17.6	18
6	17.7	17.6	17.1	17.4	17.6	17.1	17.2	17.6	18.1	18.1	19.7	22.3	22	22.5	22	21.6	20.6	21.5	20.2	18	17.2	16.7	16.5	15.5
7	15.2	14.5	13.7	13.4	13.3	13.4	13.3	16.7	20.2	21.2	22.3	23.9	24.9	24.2	25.9	25.9	26.6	25.2	22.7	20.4	22.7	23.4	23.1	22.8
8	22.8	23.2	23.3	27.6	27.8	25	26.1	29.6	30.7	31.9	34.3	37	39.1	36.4	32.9	33.7	34.8	34.4	28.4	25	24.4	22.6	19.9	20.6
9	18.3	17.5	16.6	16.1	17.4	15.4	15.5	20	21.2	23.5	21.4	22.9	23	25.1	23.8	23.6	26.8	23	24.8	20.5	18.8	18.4	17	17
10	17	16.6	16.6	18.8	21.1	22.8	24.3	27.3	30.3	31.8	33.9	33.7	28.3	27.3	27.3	27.3	23.8	23.9	25.5	23.9	24.4	20.8	19.4	18.7
11	18.6	20.1	19.1	17	16.7	16.5	15.7	20.9	21.6	21.8	22.7	25.8	25	23.7	24.3	24	22.4	21.8	19.7	19.5	18.4	16.9	16.3	15.6
12	15	14.5	16.1	16.9	15.6	14.1	13.7	17	20.5	22.3	23.2	23.8	24.7	22.6	22.9	21.5	21.1	19.9	19.2	18	16.8	16.2	15.6	15
13	15.3	14.9	14.9	14.6	14.2	13.7	14.9	16.2	17.2	19.9	20.4	20.5	20	20.4	21.3	21.3	19.7	19.9	18.2	17.3	16.5	15.6	15.1	14.6
14	16.8	17.1	17.1	17.3	17.2	17.6	17.7	18.1	20.6	20.5	22.2	22.9	22.9	23	23.3	22.5	21.5	20.8	18.7	17.9	17.5	17.3	16.7	15.7
15	15.8	15.7	15.3	14.4	14.1	14.4	14.7	17.6	19.3	19.6	21.2	22.7	23.4	24.6	23.5	23.6	24.1	23.8	21.7	20.4	18.8	18.1	17.7	19.6
16	17.3	14.7	15	13.7	13.4	12.6	13.5	16.9	23.3	27.1	26	21.6	22.1	24.9	26.5	25	24.9	23.8	21.7	19.5	18	17.6	18.2	16.3
17	15.4	15.6	14.6	12.1	12.7	13.7	13.6	13.8	14.8	17.3	19.3	21.3	21.6	20.8	20.2	19.4	18.9	18.4	17.8	17.3	17.6	18.8	18.4	18.1
18	19.8	17.2	16.5	15.8	15.8	16.9	16.9	19.8	19.8	27.2	26.8	26.3	23.7	21.9	23.4	24.8	21.6	21.1	19.1	17.6	17	16	16.3	15.8
19	15.7	14.5	14.7	14.2	14.1	14.4	14.2	16.6	19.1	20.4	21.3	21.6	22.6	22.7	22.4	21	19.8	18.5	16.9	16.2	15.9	16.1	15.8	15.8
20	14.9	14.8	14.6	13.2	13.2	12.2	11.5	15.7	18.3	20.2	20.9	21.4	22.4	22.1	21.7	20.6	20.6	19.4	17.9	17.8	17.7	16.7	15.8	15.3
21	14.2	12.6	13.2	13.2	12.9	11.5	11.3	14.3	17.1	20.7	21.1	23.1	23.5	23.7	22.9	22.4	21.1	19.2	17.6	17.4	17.5	17.8	17.8	17
22	17.1	16.5	15.7	15.9	15.3	15.8	14.9	17.6	19.1	21.6	22.1	21.9	22.1	23	22.7	21.5	21.6	20.4	18.9	18.2	18.3	18.2	16.9	16.5
23	16	16.2	15.6	15	15.3	14.6	16.5	20.9	23.1	24.6	26	28.2	29.5	31.6	30.7	32.6	33.7	32.6	29.3	29.4	28.6	27.3	27.8	26.4
24	26.5	28	30.4	30.1	29.2	24.6	22.3	23.8	25.5	25.1	28.9	35.5	36.4	28.8	29.3	29.4	28.3	23.6	21.3	20.3	20.1	19.5	19.1	17.9
25	17.8	18.2	18.1	18.2	18.1	17.9	17.4	17.6	18.5	19.6	20.5	19.6	19.4	21.4	21.4	21.9	21.2	19.9	18.5	18.1	17.2	17.1	16.6	15.7
26	15.1	14.6	14	14.2	13.5	13.1	13.2	13.9	15.7	18.4	18.1	19.7	19.3	20.1	19.7	19.5	19.2	17.2	16.2	15.9	14.9	14.3	13.6	14.1
27	12.2	13.8	14	14.4	13.3	12.6	12.7	15.6	15.6	14.8	15.7			20.1	20.5		17.5	16.5	15.4	14.3	13.6	13.1	12.6	12.3
28	12.1	11	10.2	10.2	9.4	9.4	8.3	11.9	15.1	17.9	19.4	18.8	18.5	18.6	18	17.8	16.9	16.1	15.5	15.1	14.3	13.7	12.3	12.2
29	13.4	12.6	11.9	12.7	13.4	14.3	13.8	16.7	22.5	24.5	26.1	28.1	30.1	28.6	25.2	27.6	27.3	25.7	22.5	21.8	21.1	18.3	19.2	17.9
30	17.9	15.9	16	15.8	13.4	14.9	13.8	18.1	20.1	19.2	21.1	21.3	23.6	26.4	25.7	26.6	24.6	21.7	18.9	18.1	17	18.2	17.7	18
31	16.7	15.5	13.8	13.2	11.7	11.5	11.7	15.6	21.2	23.8	22.7	22.8	23.6	24.8	25	25	25	24.6	22.4	19.9	19.5	19.1	14.2	13.5
1	14.8	13.5	14.8	13.5	16.2	14.9	13.9	17.9	24.7	25	25.8	24.2	22.2	21.6	20.9	23.3	20.1	17.7	17.7	17.1	17.4	16.4	16.9	17.5
2	17.7	18.3	19.2	19.6	20.4	21	19	18.8	19.9	21.8	23	24.7	24.7	24.2	23.8	25.6	24.4	23.4	21.4	20.5	19.3	18.8	18.2	17.8
3	17.4	16.6	16.3	16.7	17.7	18	17.9	20	24.5	26.7	28.5	29	24.4	24	24	23.9	22.7	19.8	19	18.7	17.8	17.2	17	16.1
4	14.6	16.1	17.2	16.3	16.9	17.3	16.8	18.2	20.1	21.2	20.5	19.8	16.6	16.3	17.6	18.7	18.8	18.6	17.1	16.8	15.8	15.6	15	14.3
5	13.3	12	11.8	11	11	11.2	11	12.5	15.2	17.7	19	19.3	20	19.7	19.4	18.8	17.8	16.3	15.5	14.7	12.9	11.6	12.1	11.6
6	11.3	11.3	10.1	9.9	9.6	10	8.9	11.7	14.3	17.1	19.3	19.2	19.7	20.5	21.4	19.2	17.9	18.3	16.8	16.3	14.8	14.2	15.8	15.9
7	16.1	17.9	18.9	18.8	20	19.4	19.8	21.9	24.6	26.4	27.7	29.1	30.4	31.7	33.6	27.1	27.4	26.9	25	23.2	20.5	19.4	18.5	17
8	17.1	12.5	13.2	12.7	11.9	10	9.7	12.1	15.9	18.1	19.4	18.8	20.3	19.5	18.9	19.1	18.6	17.4	15.7	15	14.3	15.2	15.3	14.9
9	15	15.2	15.1	15.2	14.8	14	14.1	14.6	15	15.8	18	18.6	17.7	18.3	18.4	18.5	16.7	15.7	15.2	14.8	14.7	14.3	14.2	13.3
10	13.1	12.3	11.8	12	10.9	10.5	9.6	10.9	13.5	17.4	19.2	18.7	19.5	19.6	19.2	18.6	17.7	16.4	15.4	14.5	14.1	14.5	14.9	14.7
11	14.6	14.8	13	13.5	15.8	13.2	14.1	15.5	22.7	25.4	27	28.4	30.8	24.6	26.6	25.8	23.9	23.2	20.4	20.3	19.7	17.8	17.8	16.7
12	16.3	16.1	15.4	12.8	11	11.6	12.5	15.4	18.5	20.8	22.7	21.1	23.5	22.7	21.4	23.5	23.4	20.2	17.7	17	15.7	15.5	14.7	12.6
13	13.4	11.8	12.1	12.5	11.8	11.8	10.9	13.1	19.3	21.4	21.9	21.4	22.6	24	22.6	23.3	22.6	21.6	21.2	21.6	20.2	17.8	17.9	16.8
14	15.1	14.9	14.4	18	15.7	15.9	17.1	17.9	22.6	25.7	25.5	27.4	27.3	30.1	28.7	28.9	26	25.2	22.6	22.9	22.5	20.3	17.5	17.1
15	16	15.5	14.3	13.9	16.1	15.9	16.1	17.1	20.2	22.3	22.7	25.7	26.1	26.8	26.5	24.9	21.7	19.4	18.4	17.1	16.7	17.6	15.6	13.2
16	14	13.9	12.8	12.6	10.9	9.8	9.3	11.4	13.3	14	15.6	17.8	17.6	17.3	16.5	15.7	16.3	16.7	15.5	15.1	15	14.3	12.1	12.8
17	14.1	13.8	13.7	14.7	14.7	14.4	13.9	13.6	14.1	15.7	19	20.5	20.2	20.1	20.4	19.7	18.3	17.3	17.5	18	17.5	17.9	16.7	15.9
18	14.8	15.4	14.9	13.7	12.8	13.2	13	14.2	16.9	19.6	19.6	21.2	21.6	22.9	22.7	21.7	20.6	20.2	19.3	17.8	17.9	17.2	16.3	15.9
19	17.6	15.6	14.3	14.5	15.9	17.9	16.1	19	22.2	27.8	29.4	28	29.5	25	22.8	21.1	20.4	19.3	16	14.2	13.2	13	12	13.2
20	12.7	13.1	13.2	13.3	11	9.9	10.3	12	12.6	13.2	16.2	18.9	16.6	19	21.1	17.5	16.5	16.7	16.4	15.8	15.2	15.4	15.6	15.1
21	14.7	14.6	15	15	15.1	15.2	14.9	15	15.9	17.3	18.6	20.7	20.8	20.9	20.8	20.6	18.9	17.2	16.4	15.7	15.5	15.1	15.4	16.1
22	14.4	13.3	14.3	13.6	13.1	13.6	13.1	14.3	16.9	20.5	19.5	21.6	25.3	22.5	22.6	21.5	20.5	19.9	18.5	17.3	16.5	16	15.9	14.7
23	13.3	15	14.1	13.7	14.1	13.4	13.6	15	17	19.7	23.5	17.3	18.8	18.7	20.3	20.4	18.3	17.2	16.7	15.7	16	15.2	14.9	14.9
24	14.8	14.7	14.8	14.6	14.4	14.5	14.5	14.7	15.2	15.7	17.8	17.7	19.2	19.2	19.8	18.7	18	16.9	16.3	16.1	16	16.1	16.3	16.6
25		20	18.7	18.2	18.2											19.4	19.6	18.9	17.3	16.3	15.7	15.2	14.8	14.6
26	14.3	13.9	11.7	11.9	11.7	11.5	10.5	10.8	14.2	17.4	17.5	17.3	16.5	17.6	18.5	19.7	18.4	17.6	15.9	15.8	14.1	14	13.8	13.4

Hours of the day – Nortier temperature data January 2002 – August 2002

Day	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*
27	13.6	13.6	13.8	14	13.4	13	12.8	12.2	12.7	13.4	14.2	14.6	15.5	17.2	18.5	18.6	17.1	14.1	13.4	13.3	13.1	12	11.3	11.8
28	11.5	11.7	11.2	11.7	11.6	11.1	11.5	12.7	14.1	15.3	17.4	18.9	20.1	20.2	19.6	19.3	18	17	15.8	16.1	16.6	16.8	17.2	17.1
29	15.6	14.8	15	15	15	15.1	14.3	14.6	15.5	15.9	18.2	17.6	18.4	18.4	18	17.5	17.1	16	14.9	15	14	12.4	12.5	11.6
30	11	10.8	10.2	10.5	10	8.6	8.9	9.9	15.6	17.9	20.7	22.4	23.5	24.5	25.5	25.8	20.4	18.5	17.7	18.8	16.7	15.9	15.4	15.8
1	15.1	14.9	12.8	10.3	9.9	9.9	9.6	10.8	14.3	16.6	18.1	18.6	18.6	19.3	19	18.9	18.3	16.2	14.4	13.4	12.7	11.9	11.8	11.9
2	11.2	10.9	10.8	10.7	10.3	10.1	10.8	11.3	14.7	17.3	18.8	19.5	19.6	20.1	19.2	18.8	18.7	17.1	15.4	15.4	14.7	13.7	13.2	13.8
3	14.5	15	15.1	15.7	15.7	15.3	14.8	14.9	15.1	16.4	18.9	16.6	16.7	17.5	16.6	16.9	17.8	16.9	16.4	15.5	14.5	13.5	13.8	13.6
4	12.8	13.9	14.6	12.8	11.3	11.8	12.9	14.6	17.3	18.9	18.6	18.7	19	19.4	18.9	18.4	17	15.9	14.8	14.6	14.2	13.9	12	10.4
5	12.2	13.8	13.9	13.7	13.9	14.1	14.1	14.3	15.2	15.4	16.8	16.8	17.7	18	18.1	17.8	15.8	14.7	14.2	14.3	13.6	13.4	13	12.8
6	12.4	12.4	11.8	10.5	11.7	11.8	11.4	11.1	12.3	13.7	14.3	15.5	15.5	15.5	15.7	15.4	14.4	13.1	11.9	10.2	9.5	8.7	8	8.2
7	7.6	6.2	7	6.5	6.8	6.9	6.1	7.4	11.1	13.8	14.6	15	15	15.8	15.8	15.4	14	12.7	11.7	10.7	10.6	10.1	9.1	9.6
8	9.1	8.7	8.5	8.8	8.9	8.5	8.7	10.4	15.4	18.7	20.4	21.2	22.5	23.8	24.6	25	23.4	19.4	17.6	16.9	16.6	15.3	12.5	13.1
9	10.9	12.7	13.1	13.6	11.1	10.7	10.9	10.5	14.9	16.4	18.4	20	20.4	20	18.6	18.4	18.3	16.4	15.1	14.4	13.7	13.1	12.9	12.1
10	11.8	10.7	10.3	9.7	10.2	9.6	8.5	9.9	13.3	15	16.1	18	18	19.9	21	19.9	17.4	15	13.7	12.8	12.3	11.9	12	10.7
11	9.3	8.2	9.6	9.4	8.5	9	8	9.1	11.9	14	15.6	15.8	17.4	19.6	19	16.9	17.6	15.8	14.7	13.9	12.3	11.9	11.6	11.3
12	9	8.3	7.8	7.1	8.2	8	8.8	9.9	11.8	14.4	17.1	17.4	18.6	18.6	18.2	16.3	15.3	14.1	12.6	12.1	11.9	12.6	12.3	11.6
13	12.9	10.3	10.1	9.8	9.6	9	8.3	9.4	12.6	15.4	17.3	18.3	18.9	18.3	18.7	18.5	17.1	16.2	14.4	13.7	14.3	14.7	15.2	14.9
14	14.4	14.4	14.5	14.4	13.7	14.1	13.9	13.6	14.7	16.5	15.9	17.8	18.2	16	15.6	16.2	17.1	17	17.5	17.9	17.7	15.9	14.5	14.7
15	14.8	13.4	13.3	11.2	11.1	11	10.4	11.2	13.7	14.7	15.7	17.9	19.6	15.9	16.5	17	16.1	14.5	13.4	12.8	12.9	12.8	12.5	12.2
16	12.6	11.9	12	12.4	12	12.1	12.7	12.5	14	15.9	17.1	18.4	18.4	19	19.2	18.4	17.2	15.7	13.8	11.7	12.1	12.1	12.5	11.6
17	11.5	12.7	9.6	10.3	11.8	11.4	10.4	11.7	14.8	19.1	20.5	20	20.3	19.6	18.5	17.7	17	15.6	14.6	13.6	13.5	13.8	13.7	12.9
18	12.4	11.4	13.3	12.2	11.5	10.6	10.5	11.7	16.1	15.9	19.7	21.2	20.9	23.2	23.7	23.4	20.9	18.1	16.9	16.3	15.2	14.1	15.2	14.9
19	14.1	14	14.1	15.2	14.7	14.8	14.3	16.4	20	26.5	28.7	30.1	31.2	33.1	34	33.8	28.6	26.5	24.4	24.2	21.3	20.4	23	23.1
20	22.6	21	22.4	22.6	22.9	22.6	22.3	23.1	25.8	27.3	28.8	30.4	31.2	33.1	33.1	32	27.7	24	21.3	20	20.5	22.1	18.4	17.1
21	17.2	18.7	20.6	19.1	21.5	22	24.3	24.7	23.7	22.4	17.4	16.7	18.5	20.3	22.3	22.8	19.1	16.2	15.5	15	14.8	14.3	14.2	13.9
22	14.3	15.2	15.5	15.9	15.1	15	15	15.2	15.7	16.6	15.5	16.5	15.5	16.7	15.5	14.6	14.2	14.3	14.5	14.7	14.4	14.2	14.3	14.4
23	14.7	14.8	14.5	14.7	14.3	14.3	13.7	13.6	14.3	15.4	15.8	16.6	17	17.7	17.5	16.8	15.9	14.2	13.4	12.9	13.2	13	12.8	13
24	12.8	11.8	11.9	11.9	10.8	11.6	12.2	12.7	13.9	14.5	15.9	16	16	17	16.8	15.3	15	14.8	14.9	14.7	15.2	15.5	15.5	14.9
25	14.7	14.5	14.5	14.5	14.4	14.2	14.2	14.3	14.6	15.1	16.2	16.9	16.7	17.3	16.9	16.1	15.7	15.4	15.3	15	14.3	14.1	14	14
26	14.1	14	14	14	12.9	13	12.9	12.4	13.8	15.3	16.5	18.1	17.2	17.7	16.6	16	14.9	14	13.2	12.6	12	12.4	11.5	10.8
27	10.7	10.7	10.6	9.8	9.8	10.1	9	12.4	11.9	13.5	18	21.9	23.9	20.5	18.5	18	17.8	16.3	14.9	15.2	14.1	13.3	12.1	12.4
28	11.4	12.2	12.4	13	13.2	13.3	13.3	13.1	13.4	14.6	15.6	16.1	15.7	15.5	15	14.8	14.3	13.9	13.7	13.7	13.6	12.6	12.5	12.2
29	12.3	12.1	11.4	11.2	11	11	10.9	11	11.1	11.7	12.4	11.4	12.1	12.6	13.1	13	12.5	11.4	10.9	9.3	9.6	9.9	9.8	9
30	8.4	8	7.8	7.5	6.2	6.4	6.2	6.1	9.4	11.7	13	14.2	14.9	14.3	14	13.7	12.8	11	9.8	8.8	8.5	8.3	7.4	7.7
31	6.5	6.6	6.6	5.5	5.8	5.9	4	5.2	8.4	11.6	12.5	14.9	15.4	15.7	16.3	15.6	14	12.1	11.9	9.1	8.6	8	8.5	9.4
1	10.5	11.1	11.2	11.1	9.7	9.1	10.4	10.9	11.1	12.4	13.7	13.6	14.9	14.9	16	15.8	15.2	13.1	11.2	10.1	8.8	8	7.3	7.8
2	6.6	8.1	7	5.2	6.4	8.4	9	9	11.9	13.9	14.9	16.3	17	17.8	17.7	14.5	14.2	11.9	11.1	11.3	12.4	11	10.3	10.6
3	11.9	10.7	9.7	11	10.5	9.7	11.6	13	16.2	18.6	20.3	21.6	22.2	22.9	23	23.1	22	18.6	17.3	16.8	15.2	15.5	15.8	17.3
4	18.2	16.9	17.1	14.3	15	16.7	17.6	17.8	19.4	21	22.6	23.8	25.6	27.2	28	28.2	27.2	23.8	22.2	20.5	21.8	20.4	20.6	21.5
5	20.9	20.6	20.8	20.2	19.3	18.5	17.6	16.8	16.4	17.9	17.4	18.6	18	17.8	17.7	17.8	15.9	14.7	13.2	12.7	12.4	13.3	13.3	13.3
6	13.1	12.9	13.2	13.4	13.3	13	13.5	13.8	13.6	13.9	14.5	15.6	16	16.2	15.7	15.1	13.9	12.5	11.3	10.7	9.4	8.9	8.6	7.8
7	6.5	7.2	7.2	7.4	6.8	6.1	6.1	7	6.9	12.6	14.2	15.7	14.8	14.3	14.4	14.1	13.1	12.2	10.3	10.1	11	12.2	12.9	13.2
8	12.1	11.9	12	11.9	12.2	12.3	11.4	11.4	11.9	13.2	13.6	15.1	15.5	15.9	15.9	15.6	14.2	13.3	12.3	11.8	11	10.4	8.9	8.7
9	8.6	8.4	9.1	8.1	8.9	8.8	9.4	9.9	11.5	14	15.4	16.7	17.4	17.4	17.5	17	15.8	14	12.6	11.2	10.6	10.3	10.6	10.3
10	8.7	10	9.7	8.5	7.9	8.4	11.4	10.9	10.8	14.1	19.8	22	23.5	24.5	25	24.9	15.7	13.7	12.9	13.2	11	9.1	8.9	10.7
11	8.8	9.2	9.6	9.9	10.4	12.1	12.9	14.8	17.2	19.9	22.5	24.3	25.5	18.2	17.9	16.3	15.1	14.2	13	12.8	13.1	13	13.1	12.7
12	13	12.4	13	12.8	12.7	12.7	12.7	12.5	12.9	12.6	12.3	12.4	12.9	15.8	13.7	13.8	13.4	12	11	10.8	10.5	11.2	11.6	11.5
13	11.6	11.6	11.4	10.8	10.3	10.5	10.6	10.5	11.5	13	13.5	14.3	14.4	14.5	14.4	14.3	14	13.3	12.5	11.9	10.9	10.9	11.4	11
14	10.6	11.2	11.5	11.5	11.2	10.6	10	10.5	11.6	13.1	14.2	15.4	15.5	16.5	16.4	16	15.3	13.6	12.1	11.4	11	11	11.3	11.1
15	10.9	10.7	12.3	12	11.9	11.4	11	9.9	11.5	13.4	15.3	16.4	15.6	16.5	16.2	15.8	15.2	14.1	13.2	12.9	12.4	12.7	12.4	10.7
16	12	12.2	10.9	11.4	10.7	9	9.2	9.3	11.8	14.4	16.4	16	16.4	15.6	14.6	13.1	12	11.6	12	12.9	11.8	11.6	11.7	8.9
17	8.4	9	9.8	10.3	11.5	9.4	9	8	8.7	10.9	12.1	13	12.9	13.3	13.4	13.2	13	12.5	11.6	11.3	12.1	12.7	12.8	12.9
18	13.1	12.7	12.7	12.9	12.5	12.2	12.2	12.3	13.1	13.5	14.1	15.1	15.1	15.7	15.2	14.2	13.6	12.8	12.5	11.9	10.8	10.6	10.5	9.8
19	9.3	9.6	9.6	9.1	8.8	8.4	8.3	7.7	9.6	12.1	14	15.6	16.3	17	17.4	17.2	16.1	13	11.4	11.1	10.3	9.8	9.5	8.4
20	8.8	8.5	7.9	7.3	8	8	6.4	8.4	12.5	14.9	16.8	18.3	19.5	20.5	21	21.3	20.3	15	13.9	14.4	16	15.1	15.9	17.6
21	17.9	17.4	17.3	16.8	17.3	16.7	17	16.8	18.1	19.8	21.1	22.6	24.2	25.6	26.5	26.3	25.6	23.1	21	19.6	19.4	19.2	19.8	18.8
22	18.7	18.2	17.4	17.4	15.6	15.6	15.5	13.7	14.5	15.4	16.1	16.8	17.5	17.8	17.2	16.2	15.4	14.3	13.1	12.1	11.7	10.7	12.1	11.7
23	10.1	8.6	7.7	8	7.4	7.8	7.1	8.1	10	12	13	14	14.2	14.5	14.7	15	14.5	13.3	12.8	12.8	12.4	12	12.8	12.2
24	11	10.8	10.4	9.9	7.9	9.4	7.8	9.5	9.5	12.7	13.6	14.7	14.7	15.6	15.3	15.2	13.9	12.1	11.1	10	8.1	8.1	8.2	7.6
25	6.9	6	6.9	6.1	7.3	7.8	5.2	6.7	10.4	13.2	16.2	18.8	17.1	17.3	17.9	17.6	14.4	12	10.8	9.3	9.9	9.1	9.1	9.6
26	8.1	9.2	8.9	8.7	9	8.8	9.2	7.9	10.1	13	13.7	14.4	12.6	12.9	13.2	12.2	11.9	10.7	9.2	8.4	8	6.9	6	6

Hours of the day – Nortier temperature data January 2002 – August 2002

Day	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*
27	6.2	5	5.2	5.4	4.7	4.5	4.6	5.3	7.3	9.7	11.6	13	13.5	13.6	12.9	12.6	11.5	10.1	9.9	10.7	10.1	7.9	7	6.9
28	8.1	7.6	7.5	7	6.5	6.3	7.1	6.4	8.5	10.3	11.7	13.4	12.6	13.5	13	12	12.3	11.9	11.7	10.9	11	10.9	10	10.4
29	10.6	9.7	10.1	10.7	10.3	10.9	10.7	10.6	10.8	11.8	12.5	14.2	14.9	15.8	15.2	14.6	13.8	12.4	11.1	9.4	9.5	9.3	8.3	7.8
30	8.7	8	8.3	8	6.6	7.2	7.1	7.6	9.6	13.3	14.8	16.4	17.5	15.6	16.2	15.7	14.6	12.8	11.8	11.7	11	9.7	9.6	9.7
1	8.5	8.3	8.8	9.4	8	8.9	8.7	9.3	12.9	15.6	14.6	16.9	18.3	18.6	19.8	20.3	17.9	16.3	14.7	13.6	12.8	12.3	11.8	9.5
2	9.7	10.8	11.5	9.2	8.9	8.6	9.9	7.8	12.2	15.5	14.4	15.7	16.7	19.7	17.3	16	15.1	13.8	13.1	12.8	13.6	13	12.9	13.2
3	13.4	14.1	14.1	12	11.1	11.8	11.8	10.2	16.3	21	22.3	23.8	24.9	26.2	23	20.7	19.5	18.9	17.6	14.6	14.5	10.1	11	11.5
4	9.7	8.9	9.6	9	8.6	6.7	6.7	7.9	8.2	10.2	11.7	12.8	15	16.7	16	18.2	16.9	14.3	13	12.8	9.9	10.4	9.7	9.2
5	8.9	8.8	9.2	8.1	8.7	8.5	8.9	9.2	9.1	10.7	11.9	12.6	14	14.8	16.1	16.7	15	14	13.7	13.2	12.9	11.9	10.3	9.8
6	9.7	9.8	8.4	8	7.2	8.8	6.5	7.1	9.9	14.4	13.3	14.6	16.2	15.9	15.9	16.3	15.9	14.5	12.7	11.6	11.4	9.3	10.6	7.5
7	8.4	8.2	8	8.4	8.7	8.5	8.2	8.3	8.7	10.8	13.6	14.7	12.9	13.7	13.7	13.4	12.6	12.3	12.1	12.1	12	11.8	11.5	11
8	11.2	11.2	11.8	12.1	12.2	12.2	12.1	12.2	12.4	12.9	13.5	13.6	14.2	13.6	13.3	13.4	12.3	12.1	12.2	12.3	12.2	12.3	12.3	12.4
9	12	12.3	12.5	12.5	12.7	12.8	13.1	12.9	12.8	13.5	14	14.9	15.1	14.8	14.7	14.3	13.2	12.1	11.1	10	10.2	8.9	8.5	8.5
10	8.1	7.4	6.6	6.1	7.4	7.6	6.2	6.3	8.8	11.6	12.7	14.4	13.7	14.1	13.9	13.6	12.8	11.6	10.4	10.4	9.9	9.4	8.9	9
11	7.8	7.8	8.2	8.6	7.6	8.3	7.8	7.9	9.6	11.9	14.2	15.4	16.4	17.4	18	17.3	15.8	13.5	12.5	12.2	11.9	11.4	10.3	9.9
12	9.8	9.5	8.7	9.5	7.1	8.7	8.8	7.1	10.3	12.1	14.8	15.2	15.8	17	16.6	16	14.3	13.6	12.9	12.4	11.8	10.8	9.5	10.6
13	10.3	10.8	9	8.9	7.2	8.1	7.9	8	8.8	10.5	11	10.9	11.6	12.5	13.1	12.5	11.8	11.4	11.2	10.5	10.4	10.5	10.9	10.9
14	11.4	11.5	11.3	11	11.1	11.1	11.1	11.2	11.7	12.3	13.6	13.9	13.8	12.9	12.3	12	11.8	11.8	12	12.1	12.1	12	11.5	11.2
15	11.1	10.5	9.4	9.3	8.5	7.5	6.7	7.1	9.3	11.5	12.1	11.9	12.7	13.4	11	12.2	11.9	11.4	10.5	9.9	9.9	9	8.2	8.3
16	7.8	7.8	7	6.4	5.8	5.5	5.8	5	7.8	10.2	11.1	11.7	12	12.1	12.5	12.1	11.3	10.3	9.4	7.9	7.8	6.4	5.8	5.1
17	6.6	4.7	5	4	3.9	4.1	3	4	6.3	10.7	12	13.6	13.9	13.6	13.2	13.5	13.5	11.6	10.3	9.8	9.4	8.6	7.5	9.1
18	7.6	8.4	7.4	6.2	5.7	6.7	6.7	8.6	12.6	15	16.1	17.5	18.4	15.4	15.7	15.2	14.2	12.4	10.8	10.4	10.7	10.4	11.5	11.3
19	11.6	10.8	11.8	10.7	10.6	9.9	10.5	11.7	14.6	15.9	16	17.3	17.9	17.9	19.7	18.6	15.8	12.8	11.4	11.1	11.6	11.6	11.8	11.2
20	11.8	13.5	13.8	13.8	13.7	14.1	14.5	14.5	17.6	18.7	19.6	21	21.5	22.2	22.3	22.3	21.6	18.3	17	15.2	14.6	13.7	12.3	12.9
21	13.1	12.4	13.8	14.8	14	12.8	14.8	14.9	17.1	19	20.5	21.2	21.9	23.2	23.8	23.8	23	19	17.3	15.9	15.8	18	17.3	16.3
22	15.2	16.3	16.9	16.1	15.4	16	15.8	14.9	14.7	17.1	14.9	14.5	15.5	15	13.7	14	12.2	11.6	10.9	11	11	11.2	11.2	11.3
23	11.2	10.8	10.2	10.1	10	9.6	9	9.3	10	11.2	11.6	12.7	13.5	12	11.6	12.3	12.1	12	12	12	11.8	11.7	11.9	12.2
24	12.2	12.1	12.2	12.2	12.3	12.2	12.3	12.1	12.4	13	14.1	14.9	15.8	16.1	15.9	15.1	12.8	12.6	11.9	11.6	11.6	11.3	11.4	11.5
25	11.5	11.1	11.2	11.2	11	11.2	11.4	11.5	11.6	11.6	12.2	12.9	13.7	14.8	14.2	14.1	13.1	12.4	11.7	10.7	9.7	8.6	8.5	8.5
26	7.8	7	6.8	5.8	5.8	6.2	5.2	6.9	9.9	13	13.8	15.1	15.7	15.9	15.1	15	14.2	12.5	10.6	9.7	8.8	9.1	10	10.1
27	8.9	9	9.9	11.7	12.9	13.3	12	12	12.3	12.8	13.5	13.9	14.4	15	15	14.9	14	12.6	11.2	10.2	10.7	10.1	9.4	8.9
28	8.7	8.4	9.1	9.4	8.3	7.8	7.5	8	9.6	12.7	14.4	15.8	16.8	15	14.2	14.1	12.8	12.8	12.6	12.6	12.4	12.9	12.9	13.7
29	14.2	13.9	13.7	13.2	12.8	12.8	12.8	12.8	12	13.2	13.9	13.2	14.3	15.5	15.3	15.4	14.2	12.7	11.8	10.9	10.8	10.4	10	10.2
30	10.4	9.9	9.7	10.6	10.2	9.8	8.3	8.1	11.1	11.4	12.7	13	13.5	13.5	13.4	13.4	12.5	10.4	8.6	7.9	7.6	7.2	6.7	6.6
31	5.7	5.3	6.7	6	7.8	8	7.7	9.7	11.5	13	14.5	15.8	16.7	17.5	18.1	18.1	17.5	16.2	16.1	15.8	15	14.7	15.2	14.3
1	15.1	16.2	16.8	16.6	17.3	16.6	14.2	13	15.1	16.1	15.2	15.6	15.8	15	14.4	14.3	14.1	13.1	13.3	13.6	13.7	13.6	12.9	13.1
2	12.3	12.5	12.1	11.3	11	10.9	10.7	10.5	11.8	13.6	14.2	15	15.8	15.8	13.4	14.7	14.2	12.7	10.9	10.4	11.3	10.7	11.1	10.9
3	10.8	10.4	10	9.6	9.5	9.3	8.7	8	10.1	11.8	13.3	14.6	14.9	14.2	14.8	14.3	13.7	12.5	10.8	9.3	9.2	9.4	8.9	7.9
4	6.6	7.1	6.3	5.8	5.3	5.9	6.1	6.7	10.9	13.5	15.1	16.8	17.6	18.3	18.6	16.4	15.6	13.1	10.6	10.1	9.6	9.2	8.8	8.8
5	8.2	8.2	8.7	8.3	8.9	8.1	6.2	8.6	12	15.9	16.2	15	15.1	16	15.4	15.1	13.9	11.8	10.4	9.9	8.4	8.6	8.2	7.6
6	7.1	7	7.2	7.3	7	4.8	3.8	5.2	9.5	11.6	13.9	14.7	14.9	15.8	16.9	15.5	14.8	13.5	12	11.4	10.8	10.1	9.5	9.7
7	10.1	10.5	7.7	9.5	9.4	9.3	9.9	10.8	13	19.1	21.7	22.7	23.5	17.4	18.9	20.4	18.2	16.8	15.1	15.9	14.7	14.8	14.9	13.9
8	13.3	12	12.8	15.4	11.9	8.4	11.3	14.2	16.1	18.1	19.6	19.2	20	20.8	21.4	21.6	20.5	17.8	16.5	13.8	15.9	15.1	13	12.8
9	13.1	13.1	13.6	13.4	12.7	12.8	13	15.9	16.9	18.2	23.1	21	23.4	24.4	18.2	18.7	20	15.9	15.3	14.7	12.3	13.1	11.8	14.1
10	15.3	15.1	16	14	15.2	14.4	17.1	17.4	20.9	25	27	28.2	30.3	32	32.6	32	29.7	26.9	23.4	22.8	21.7	22.1	20.7	19.9
11	19.4	18.9	18.7	18.8	18.9	19.7	20.2	19.5	21.2	23.9	26.9	30.7	24.1	20.5	18.7	17.8	16.7	15.4	15.3	15.1	15	14.8	14	13.8
12	13.3	13	13	12.6	12.7	12.5	11.8	11.7	13.5	14.7	15.3	16	16.3	16.1	16	15.4	14.4	13	11.6	10.2	9.7	10.3	11	11
13	10.1	10.5	9.5	9.8	9.5	9.6	10.1	10.5	11	12.8	13.1	14.2	14.3	14.2	14.3	14.1	13.3	12.1	10.9	8.9	8	7.8	6.5	6.9
14	4.9	6.2	6.4	6.8	4.9	3.9	3.9	5.9	9.4	11.5	13.2	13.1	14.1	13.5	13.8	13.6	13.2	11.6	11.4	10.9	9.9	9.3	8.5	8.4
15	7.3	5.9	6.8	5.6	5.9	6.2	5.4	6.5	9.3	11.8	12.5	13.7	15	15.7	15.6	15.7	15.4	13.8	12.3	11.7	10.6	9.2	9.2	9.3
16	9.3	8.7	8	7	6.2	6.5	6.1	6.9	11.8	13.6	16.2	15.8	16.2	16.7	17	16.9	20.1	18.2	15.1	14.3	13.5	12.3	11.1	10.8
17	10.9	9.9	10.7	9.5	10.9	10.2	9.6	10.2	13.7	16.8	16.8	16.7	17.6	17.7	18.6	18.2	17.3	17.3	15	14.2	12.4	11.8	11.6	10.7
18	10.6	10.3	9.9	9.5	7.8	9.3	8.8	9.1	14.8	15.4	17	16.9	17.4	18	17.4	16.7	15	14.1	12.7	11.7	11.1	10.7	11	10.5
19	12.4	10.8	11.5	12.1	11.9	10.4	11.7	14.3	18.3	20.3	21.6	22.5	23.7	22.9	19.6	19.6	18.7	16.9	15.6	15.9	16.6	15.7	15.4	15.9
20	16.4	16.7	18.2	17.9	18.5	18.7	18.4	18.6	21.6	22.8	24.3	26.5	28.5	30	31	31.5	27.1	23.4	21.4	18	16.8	16.8	15.2	17.8
21	15.1	14	13.9	11.1	12.8	11	10.5	10.3	10.9	12.3	13.8	14.6	15.2	13.6	14.3	13.7	12.9	12.2	11.6	11.7	11.2	11	10.6	10.9
22	11.3	11.7	11.3	11.1	11.5	11.3	11.6	12.5	12.8	12.8	14	15.2	15	15.3	15.4	17	17.6	17	16.3	16	17.2	17.9	17.6	17
23	16.3	15.4	14.7	14.9	15.5	15.5	14.3	14.6	14.1	14.8	14.4	14.7	15	14.6	15.1	14.5	15.3	14.6	12	11.8	11.4	12.5	12.5	12.8
24	12.5	12.4	12.4	11.5	10.7	11.8	11.8	12.1	12.8	13.7	14.6	15	15.9	15.5	15.3	14.9	14.5	13.3	11.5	11.2	11.3	10.6	9.8	9.3
25	9.7	9.7	8.2	7.8	7.1	7.2	7.5	8.9	11.8	13.2	14.5	15.2	15.9	16	16.3	16.2	15.3	14.3	12.9	12	9.9	8.7	8.2	9.3
26	10.1	10.9	11.3	11.8	11.9	12	11.8	12.1	12.7	13.8	13.6	15.1	15.9	15.8	15.6	15.9	15	14.3	13.4	12.6	10.1	9.3	8.7	7.4

Hours of the day – Nortier temperature data January 2002 – August 2002

Day	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*
27	8.4	8.1	6.9	8	9.4	9.5	8.6	8.6	13.2	14.6	15.4	15.3	15.9	16.1	16	15.7	14.7	13.4	11.8	10.4	9.5	8.4	9	8.4
28	8.2	7.5	6.5	7	7.9	7.7	7.6	8.7	13.2	15.2	16	16.6	17.2	16.9	16.6	16.6	15.6	14.5	13.2	10.9	9.9	8.9	11	10.7
29	10.9	11.4	11.5	11.3	11.8	11.9	12.2	12.6	14.7	16.4	16.7	17.3	16.7	17.3	16.9	15.6	15.2	14.4	14.1	13.2	10.9	10.2	9.6	9.4
30	9.4	9.3	9.4	9.3	9.2	7.8	7.9	9.5	11.5	13.5	10.9	14.1	14.6	15.3	12.6	12.1	12.5	11.1	9.8	7.1	7.3	6.1	6.3	6.1
31	6.4	6.2	6.2	6	5.5	5.9	5.9	7.8	10.2	13.4	14.5	16.1	15.7	15.5	15.7	14.9	14.1	12.9	10.4	9.1	8.8	8.2	7.4	7.8

Table B. 1: Nortier temperature data

B.2 Nortier Humidity Data

Hours of the day – Nortier humidity data January 2002 – August 2002

Day	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*
1	94	95	95	95	94	96	95	82	75	68	65	64	58	60	61	60	63	66	67	70	70	74	74	75
2	76	75	81	81	81	85	73	63	59	55	45	44	45	42	40	37	39	32	41	47	50	50	50	59
3	74	85	92	94	95	95	91	85	81	73	76	58	60	54	53	40	62	63	65	72	80	83	84	86
4	88	90	93	94	95	95	95	85	65	68	67	71	68	64	60	52	66	71	78	75	76	81	90	93
5	94	94	95	95	95	95	96	93	86	76	69	64	61	61	65	68	68	70	78	86	88	89	89	89
6	88	89	94	95	95	95	96	96	88	78	79	75	73	69	61	61	57	52	59	77	84	85	88	90
7	90	91	92	94	93	94	94	89	77	65	60	65	66	58	59	59	62	61	67	74	76	79	72	76
8	76	79	82	85	83	81	71	60	54	51	53	46	52	39	42	42	44	44	45	53	60	66	69	75
9	81	83	88	89	87	90	90	78	65	65	61	62	56	55	58	56	62	72	80	86	89	91	92	93
10	93	92	92	93	92	91	91	81	68	59	57	61	62	55	55	50	47	49	59	71	74	82	84	88
11	87	90	92	78	72	71	66	57	52	69	73	80	81	68	74	73	80	89	81	87	88	90	87	88
12	89	88	93	94	94	94	94	86	73	66	56	52	53	52	54	49	52	56	61	71	74	78	82	75
13	74	77	78	78	83	84	73	57	44	51	48	46	40	47	45	47	47	56	57	61	68	74	81	86
14	89	90	89	91	92	92	95	77	69	63	61	59	62	62	62	62	65	68	74	79	82	84	87	89
15	91	93	93	95	95	95	96	96	82	74	70	72	71	66	63	66	70	71	78	85	87	88	89	89
16	90	91	92	94	93	94	95	94	92	83	67	50	56	62	60	61	69	70	76	86	88	84	94	94
17	94	95	94	87	90	92	93	86	86	79	77	90	84	79	90	92	82	82	79	85	90	93	94	95
18	95	94	94	94	95	96	96	88	73	75	71	71	69	71	73	70	71	71	69	71	67	63	64	67
19	71	76	76	80	81	85	83	80	68	62	61	59	65	58	59	59	65	67	76	79	80	81	83	85
20	90	89	89	90	91	92	93	85	76	70	70	66	64	60	65	67	69	71	77	82	83	83	82	82
21	85	88	91	93	93	92	91	76	65	67	62	61	58	62	60	65	68	70	79	84	84	86	87	89
22	91	93	94	95	95	96	96	93	80	74	69	66	64	65	57	67	74	77	80	81	82	81	81	81
23	82	83	85	89	92	91	92	76	72	73	65	64	64	66	68	67	68	65	68	74	76	79	80	83
24	85	86	86	88	89	90	90	78	62	64	62	60	56	54	51	47	50	54	58	62	64	72	77	81
25	83	83	86	85	88	84	81	74	75	85	83	78	77	76	71	74	67	74	86	89	93	94	95	96
26	95	95	94	95	95	96	96	90	77	69	65	63	63	66	66	66	71	75	81	87	90	90	91	91
27	90	88	88	85	84	83	82	73	70	67	66	69	69	73	76	83	84	84	85	88	87	89	91	92
28	92	91	85	87	90	91	88	71	64	57	60	59	66	70	71	70	75	75	81	85	86	87	88	90
29	91	89	90	90	92	94	90	82	73	70	71	73	71	72	70	69	74	78	85	88	91	91	92	93
30	92	93	92	93	93	94	93	90	80	80	69	64	65	63	62	57	61	62	61	63	71	76	69	71
31	78	80	81	80	80	79	73	54	32	47	43	39	41	33	34	44	42	42	37	46	47	46	50	51
1	46	45	41	37	40	38	34	30	30	27	21	16	29	26	22	30	35	30	34	39	44	35	36	37
2	40	44	47	49	50	51	44	42	39	44	44	50	52	44	49	56	52	53	53	75	81	84	88	85
3	92	94	95	95	95	94	94	89	79	76	76	68	68	72	63	66	63	70	72	79	82	83	86	90
4	91	92	93	93	93	95	96	95	88	75	67	62	56	54	56	59	55	57	64	69	67	69	68	72
5	77	82	86	90	92	93	94	82	76	73	69	68	62	62	64	62	68	71	69	83	86	87	88	86
6	91	92	92	91	93	94	95	96	96	96	94	73	56	57	47	56	63	63	64	71	74	71	67	66
7	71	74	73	80	81	82	77	56	41	30	35	33	32	33	37	32	35	41	50	58	59	62	58	56
8	51	51	48	48	46	44	40	27	23	32	26	25	22	26	27	24	26	29	31	40	37	38	36	36
9	41	42	36	41	51	66	49	32	26	24	24	23	30	39	55	57	65	74	88	92	95	95	96	95
10	93	89	87	86	85	84	83	81	78	76	70	66	63	62	63	66	73	78	81	84	85	88	88	89
11	92	94	95	96	96	96	95	95	94	92	82	73	73	74	62	61	55	57	57	61	68	70	76	80
12	80	83	84	86	87	87	89	77	68	68	59	58	56	54	53	57	60	63	71	69	73	74	78	82
13	83	86	87	89	90	90	92	83	73	66	61	63	70	65	63	64	67	73	83	88	90	91	91	91
14	93	94	95	93	94	95	95	93	81	68	66	63	64	66	62	63	64	72	79	87	90	91	91	92
15	93	92	90	88	84	80	79	76	71	72	90	94	87	75	75	73	70	68	71	79	78	81	82	81
16	83	85	86	87	88	89	89	68	54	54	54	51	44	48	47	47	48	50	57	64	66	72	75	75
17	73	71	71	68	70	63	51	33	34	31	40	31	29	28	35	31	32	30	33	39	46	41	31	33
18	32	32	33	35	28	33	30	25	20	16	14	12	11	25	26	18	20	33	63	79	83	80	80	77
19	72	74	72	67	55	42	37	33	26	30	39	45	34	37	46	52	44	44	38	45	69	78	79	76
20	64	82	72	71	81	94	83	70	70	48	71	76	31	36	54	43	43	41	40	50	52	49	48	49
21	56	68	76	72	77	79	78	73	77	68	51	42	39	38	42	38	43	42	57	59	54	68	81	86
22	90	91	90	93	94	94	95	96	95	91	85	79	71	69	72	77	81	91	93	93	92	92	91	90
23	88	89	90	91	93	95	95	95	77	76	78	73	77	64	71	74	81	87	93	84	79	74	38	38
24	71	69	66	77	82	85	86	51	42	38	22	38	45	37	47	40	43	46	44	40	60	61	56	53

Hours of the day – Nortier humidity data January 2002 – August 2002

Day	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*
25	71	83	83	82	73	60	70	65	50	41	42	32	34	34	31	38	45	45	49	54	56	59	59	53
26	65	72	70	69	63	49	55	42	31	28	27	24	21	21	33	29	46	43	46	44	46	45	59	69
27	82	84	83	78	83	82	82	74	73	48	65	56	68	53	74	70	61	56	57	73	71	69	69	69
28	67	81	82	80	74	73	86	74	78	94	91	85	70	54	66	66	68	60	82	86	86	91	93	95
1	95	95	95	95	95	95	94	91	86	77	69	67	69	71	69	68	71	77	79	85	88	90	91	93
2	93	95	95	96	96	97	96	97	96	97	86	79	72	74	73	72	74	75	82	84	78	77	78	77
3	85	88	87	86	80	80	81	77	64	67	59	54	50	53	55	57	56	56	61	68	70	73	78	79
4	80	83	83	83	85	85	83	78	59	52	48	42	41	61	60	67	71	75	70	80	81	83	90	87
5	91	91	93	92	87	91	86	65	63	67	75	66	68	55	67	75	78	78	83	86	90	92	93	94
6	94	95	95	95	96	96	96	96	96	96	90	71	68	66	65	66	74	67	71	79	79	75	76	78
7	80	82	84	84	84	86	88	78	64	60	56	49	44	47	43	43	41	45	52	60	52	50	52	53
8	53	51	51	41	40	47	45	36	34	32	29	25	22	25	30	29	26	26	38	47	49	56	64	62
9	72	77	83	85	79	89	88	70	68	61	69	61	58	51	56	58	47	61	56	72	77	79	87	87
10	87	89	90	80	66	60	56	49	41	37	33	34	48	50	47	46	61	59	54	59	57	71	78	82
11	86	79	81	90	91	91	91	76	71	69	67	59	65	64	60	60	62	64	70	68	72	79	83	86
12	88	86	83	78	82	89	92	85	71	62	59	57	54	62	60	63	63	67	68	73	80	81	85	85
13	87	89	92	94	95	95	96	96	95	78	75	75	76	74	71	72	77	77	84	88	89	91	92	91
14	94	94	94	94	94	94	93	91	80	79	73	67	66	67	68	73	76	78	90	93	94	95	95	93
15	95	95	95	96	96	96	96	93	80	78	72	64	60	55	58	51	42	35	49	50	57	62	53	40
16	50	72	84	86	79	87	93	83	48	34	49	62	58	51	46	48	44	46	51	66	74	69	68	82
17	88	94	94	93	95	96	96	96	96	95	83	75	72	73	71	73	75	79	82	84	82	79	80	82
18	74	82	87	90	92	87	86	76	74	41	48	50	60	67	61	56	69	73	82	88	88	91	91	91
19	92	93	93	93	93	93	92	87	77	65	59	56	52	54	56	63	68	73	80	84	84	82	82	82
20	85	85	84	85	87	90	88	81	71	63	61	59	58	59	62	65	65	69	73	74	74	78	83	85
21	87	86	88	89	89	90	90	87	78	63	62	60	60	58	60	62	67	75	82	87	87	89	90	90
22	91	91	94	94	94	94	93	91	82	73	68	72	72	70	66	66	62	62	65	71	74	74	78	80
23	82	81	80	82	83	83	81	64	59	50	46	39	35	31	33	27	24	26	32	32	33	35	33	33
24	30	25	20	22	26	33	42	38	37	42	29	14	13	29	27	28	33	53	67	74	83	89	90	92
25	94	93	93	93	93	94	94	95	93	84	79	81	83	74	72	68	64	64	69	70	72	74	78	81
26	84	86	85	86	87	91	92	91	83	73	74	70	70	66	64	64	62	64	71	71	77	79	80	82
27	82	85	83	83	86	90	90	81	81	94	92		74	73	68	70	76	77	80	80	79	82	83	84
28	83	84	86	88	89	87	89	82	64	49	40	50	53	55	60	61	64	69	69	67	65	60	65	66
29	62	66	67	64	58	47	48	40	25	22	19	17	14	22	28	21	21	23	32	31	31	36	32	34
30	34	39	38	39	45	41	44	37	34	44	44	52	45	34	38	34	42	48	60	60	64	56	58	57
31	60	62	64	84	90	90	92	82	52	43	46	48	41	36	35	34	32	33	36	39	39	40	78	82
1	82	82	82	75	54	61	60	47	32	31	30	33	57	52	64	46	74	87	90	92	94	94	94	93
2	93	90	83	79	73	68	82	85	83	77	73	62	62	63	64	58	62	65	71	75	79	81	83	84
3	87	88	90	90	88	88	87	84	64	49	44	42	59	62	63	61	68	81	82	80	81	81	81	90
4	89	89	91	91	90	89	90	89	79	75	76	82	86	91	80	71	69	71	78	80	82	88	92	93
5	94	95	95	95	96	96	96	96	96	71	62	64	54	56	56	62	63	68	66	64	64	68	66	69
6	70	73	77	75	74	68	73	68	59	41	32	45	40	36	31	46	56	55	59	55	58	53	41	34
7	31	25	23	23	21	21	22	19	16	15	13	12	12	12	10	23	16	14	14	18	27	26	26	27
8	27	54	73	88	91	91	91	91	84	73	70	73	65	71	75	73	76	80	87	89	90	91	91	93
9	92	92	92	90	91	93	95	96	95	93	82	76	80	74	73	76	85	91	93	88	88	90	91	90
10	92	93	94	95	95	95	95	96	92	62	56	58	52	52	55	57	62	65	68	76	72	68	65	66
11	66	65	71	67	55	63	65	62	38	29	27	24	19	38	32	33	36	38	48	45	45	50	49	51
12	54	54	57	66	81	73	67	60	53	50	38	47	40	50	61	52	51	61	72	77	84	85	87	89
13	89	93	93	95	95	95	96	91	62	57	54	56	48	35	41	51	54	53	53	52	54	62	63	71
14	80	80	68	49	60	57	50	52	46	33	29	31	30	23	24	23	31	33	39	39	37	38	66	71
15	76	79	75	77	70	70	67	69	56	50	52	40	39	35	37	50	67	78	75	80	78	77	87	93
16	94	95	95	95	95	95	95	96	95	94	84	74	76	79	83	89	88	86	86	86	89	93	94	96
17	96	96	96	96	96	96	96	96	96	96	77	70	71	73	73	80	86	88	85	86	87	89	90	92
18	93	93	93	95	95	94	96	96	94	79	74	65	63	57	55	61	64	61	63	65	65	66	67	67
19	53	55	54	53	42	40	43	34	35	19	17	28	23	39	51	60	64	67	82	89	94	95	96	96
20	96	96	96	96	96	96	91	88	91	93	82	69	79	62	56	79	85	85	87	87	91	91	91	88
21	93	94	95	95	95	96	96	96	96	92	82	70	68	70	70	72	78	87	90	91	92	91	92	92
22	93	95	95	95	95	94	95	93	88	67	73	65	52	62	61	65	64	66	72	75	80	78	75	80
23	84	81	81	84	84	88	90	90	87	77	66	83	78	78	72	70	80	87	88	92	90	93	93	93
24	94	95	96	96	96	96	96	96	95	92	83	80	75	75	74	80	83	88	91	92	93	93	93	
25		70	79	80	80											78	75	79	85	89	92	93	94	93
26	94	94	94	95	95	95	95	96	97	93	80	79	83	79	74	70	77	79	87	88	90	92	94	95

Hours of the day – Nortier humidity data January 2002 – August 2002

Day	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*
27	96	96	96	96	96	96	96	96	96	97	96	97	96	86	77	75	81	87	93	94	95	96	96	96
28	96	96	96	96	97	97	97	97	97	96	83	70	64	61	64	66	71	74	81	81	77	74	71	73
29	85	93	95	95	96	95	96	95	95	93	80	84	77	69	75	79	75	75	79	74	76	81	81	83
30	84	85	86	85	86	86	86	83	60	54	40	33	31	28	26	27	47	51	54	42	51	55	58	57
1	57	60	72	86	90	92	93	94	91	82	73	74	74	68	70	70	70	81	89	93	94	94	94	93
2	93	92	94	94	94	93	94	93	90	73	68	64	62	60	63	65	67	71	80	78	79	81	85	85
3	86	87	92	93	94	94	95	95	96	92	77	86	86	83	84	84	78	79	74	80	79	81	80	81
4	83	82	78	83	84	81	79	76	66	62	68	71	68	66	73	78	84	87	90	91	92	93	93	93
5	95	95	96	96	96	96	96	96	96	94	89	87	82	80	82	74	76	81	87	88	87	87	89	89
6	89	91	91	92	90	93	92	93	93	81	69	60	64	60	58	57	62	65	72	76	77	80	82	81
7	83	86	87	87	86	84	83	82	72	57	51	56	52	56	57	63	74	80	86	88	78	77	78	77
8	76	78	79	77	74	72	68	63	48	31	27	25	23	21	18	17	27	30	34	38	32	36	50	50
9	71	57	51	46	57	63	58	61	45	44	44	51	48	54	66	69	70	79	84	86	88	91	90	91
10	93	92	93	94	94	95	94	94	91	82	73	61	61	52	50	53	62	69	75	78	78	73	75	84
11	85	85	84	84	86	87	89	90	82	72	65	70	67	57	58	67	63	69	75	79	82	86	85	84
12	85	89	90	93	93	92	92	91	90	81	67	68	62	63	65	76	81	84	87	88	90	90	88	88
13	87	89	91	93	93	93	93	93	90	76	69	68	64	68	68	71	78	81	86	89	91	91	91	90
14	92	94	95	95	95	95	95	95	96	91	90	80	77	83	84	81	71	71	65	66	70	77	81	78
15	77	82	86	91	93	90	93	92	88	87	85	73	65	84	84	84	88	93	94	95	95	96	96	96
16	96	96	96	96	96	96	97	97	96	91	75	67	68	66	62	67	72	78	86	88	90	88	89	88
17	88	88	88	82	75	74	80	72	65	44	32	42	48	62	72	76	79	84	88	88	85	85	86	88
18	87	87	82	81	84	85	87	87	78	77	64	61	62	50	47	47	54	68	72	69	75	72	65	67
19	70	70	75	63	56	57	57	47	41	24	18	15	14	12	10	10	19	20	18	17	21	23	20	20
20	21	23	22	23	23	24	24	25	23	20	18	15	14	13	13	14	20	28	30	32	30	29	34	37
21	37	32	31	36	32	33	31	31	40	56	65	77	78	61	55	56	61	80	86	87	85	86	88	88
22	87	83	82	84	87	90	92	92	92	88	92	87	91	87	91	94	95	96	96	96	96	97	97	97
23	97	97	97	97	97	97	97	97	97	97	96	87	79	73	68	73	77	84	91	93	94	94	95	95
24	96	96	96	96	96	96	96	97	96	94	86	87	86	81	82	90	90	90	88	90	87	86	88	94
25	95	96	96	96	96	96	96	96	96	96	95	89	89	86	88	89	91	93	94	95	95	96	96	96
26	96	96	96	96	96	96	96	96	96	93	76	72	78	76	81	82	83	87	90	90	90	85	86	89
27	88	85	85	85	85	81	84	69	85	82	58	42	37	60	59	61	59	68	73	71	76	81	91	93
28	94	94	95	96	96	96	96	95	94	87	79	75	77	82	86	87	91	94	95	96	96	96	96	96
29	96	96	96	96	95	94	94	94	94	94	94	93	90	84	77	76	77	84	87	90	91	93	90	89
30	90	92	93	93	93	92	91	90	78	68	56	47	56	62	62	68	72	77	82	86	87	88	89	87
31	88	88	85	85	81	82	84	83	80	73	59	51	60	54	48	40	55	67	68	77	77	77	75	76
1	75	78	79	78	90	91	93	92	91	79	64	63	58	58	51	51	56	65	72	71	73	73	76	75
2	81	69	72	79	75	64	59	58	45	39	35	34	29	25	46	52	58	61	59	49	38	43	44	44
3	41	46	51	45	47	50	44	41	35	31	30	28	29	28	28	29	32	42	44	44	50	47	45	40
4	36	40	39	48	46	40	37	36	33	32	31	31	30	29	26	26	26	34	37	43	39	42	42	39
5	41	41	40	42	44	46	49	52	55	49	58	66	73	78	77	73	84	88	91	94	95	95	95	95
6	95	96	96	95	95	95	95	95	95	93	90	78	69	67	68	72	76	82	83	83	83	85	81	83
7	85	86	86	86	86	85	86	83	86	75	60	64	70	66	62	59	68	74	83	84	76	73	72	73
8	92	94	95	95	95	95	95	95	96	95	95	86	80	78	73	78	84	88	93	94	95	95	95	95
9	95	95	96	95	94	93	92	91	88	78	69	61	57	59	59	61	66	73	78	85	85	85	84	85
10	88	83	85	80	81	80	64	61	71	66	33	26	25	25	21	22	69	74	73	72	76	81	81	86
11	88	92	78	68	65	58	54	46	41	33	26	23	32	56	61	68	72	84	85	86	92	93	94	94
12	93	93	93	93	94	94	94	93	93	92	90	88	90	76	84	83	80	85	86	88	88	86	86	86
13	86	87	90	92	92	91	92	93	90	84	79	77	78	76	81	81	84	87	89	90	91	92	92	93
14	93	94	95	94	94	95	95	95	96	91	86	73	72	68	68	71	76	83	89	88	89	89	90	91
15	91	90	85	86	84	88	89	88	90	87	77	76	75	71	72	75	79	83	86	85	88	88	88	91
16	93	94	94	94	94	94	94	93	91	79	71	75	76	79	85	90	94	94	92	87	91	92	94	95
17	94	96	95	94	93	93	94	95	96	96	96	95	92	89	86	86	87	89	91	92	92	92	94	95
18	96	96	95	95	95	96	96	96	96	96	96	95	93	90	90	90	92	93	94	95	95	96	96	95
19	96	96	96	95	95	94	93	94	92	85	74	56	51	45	45	39	51	71	75	78	80	85	86	87
20	86	84	83	82	73	71	76	71	53	45	36	33	26	22	21	20	22	55	55	35	29	32	28	24
21	25	26	27	27	27	29	29	30	29	28	27	26	24	24	23	25	27	30	34	38	38	39	39	42
22	41	43	44	43	57	59	59	66	76	80	80	79	76	71	71	69	72	79	82	81	84	85	86	89
23	91	90	86	83	81	78	82	83	83	80	85	81	83	82	83	83	85	89	91	92	94	94	95	94
24	94	95	94	94	94	95	95	95	96	93	89	83	78	74	71	65	71	83	87	88	90	91	90	92
25	93	93	92	91	92	91	93	91	88	80	61	44	61	63	60	61	73	81	85	84	74	70	65	65
26	68	65	86	75	65	85	86	90	92	84	76	72	91	94	95	93	90	88	90	91	91	90	92	91

Hours of the day – Nortier humidity data January 2002 – August 2002

Day	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
27	92	92	90	86	86	85	87	81	76	73	59	53	47	55	63	61	63	70	78	78	80	85	87	89
28	92	92	92	94	93	94	93	92	88	82	78	73	78	75	79	85	85	89	90	91	93	93	94	95
29	95	95	95	95	95	95	95	95	95	95	95	89	79	76	78	82	85	89	92	93	94	95	95	95
30	95	95	94	93	92	92	91	92	90	84	70	64	53	73	72	75	81	87	89	86	85	84	86	83
1	85	84	78	71	78	74	78	69	62	54	71	59	48	63	52	48	61	60	53	59	60	58	55	66
2	65	56	55	60	60	60	54	65	53	49	60	64	52	40	56	72	77	80	77	69	55	55	57	50
3	48	46	38	44	49	44	43	50	41	28	26	22	22	21	39	41	46	39	38	52	66	79	86	89
4	91	93	92	94	94	95	95	95	96	95	95	85	78	74	76	67	66	79	86	89	91	93	93	95
5	95	95	95	95	96	96	96	96	96	96	95	92	82	78	76	75	80	82	76	74	76	79	90	91
6	90	91	92	89	89	92	94	93	87	79	81	75	68	69	67	69	71	74	72	75	77	83	93	93
7	95	95	95	96	96	96	96	96	96	97	94	90	91	90	88	90	91	93	94	95	95	95	96	96
8	96	96	96	96	96	96	96	96	96	96	96	96	94	93	93	91	94	95	95	96	96	96	96	96
9	96	97	97	97	97	97	97	97	96	92	83	79	80	80	73	77	80	85	88	90	91	90	90	89
10	88	86	88	92	91	92	92	92	92	84	75	63	76	76	75	74	78	86	88	90	91	91	91	87
11	88	91	87	85	85	82	85	86	83	77	67	73	62	65	61	60	59	74	72	76	76	81	83	87
12	87	87	89	90	92	91	92	92	91	85	75	76	79	76	79	79	85	88	90	92	93	93	93	94
13	93	94	94	95	95	96	96	96	96	96	96	96	96	96	93	92	94	95	95	95	96	96	96	96
14	96	96	96	96	96	96	95	94	93	90	83	79	78	85	91	93	95	95	95	96	96	96	95	95
15	94	94	94	93	92	90	89	89	84	81	73	77	76	73	85	79	82	84	85	84	87	88	88	89
16	91	91	90	91	91	91	90	91	89	81	71	72	68	68	69	70	72	77	79	81	83	84	85	87
17	85	88	83	85	85	83	85	83	79	64	59	55	63	56	63	62	61	74	79	76	77	77	85	75
18	78	76	77	84	82	80	79	78	65	58	55	50	48	69	68	62	56	64	67	73	74	76	74	69
19	62	63	59	62	63	65	63	56	43	38	39	38	35	32	26	38	57	58	65	67	62	61	51	62
20	63	57	56	56	57	57	56	56	47	44	41	37	35	33	33	32	33	45	45	51	55	58	60	57
21	58	62	58	54	56	59	51	50	45	39	37	36	35	33	32	32	33	49	50	52	50	40	42	44
22	46	35	32	35	37	36	37	40	44	39	66	70	67	61	73	74	84	88	89	90	88	86	86	85
23	84	83	85	85	84	84	86	86	85	79	83	75	73	93	94	93	92	93	94	94	95	95	96	96
24	96	96	96	96	96	96	96	96	97	97	96	93	87	76	80	82	93	94	95	96	96	96	96	97
25	97	97	97	97	97	97	97	97	97	97	96	96	91	87	87	86	90	91	93	94	95	96	95	96
26	95	95	95	95	95	96	96	96	95	91	79	69	67	66	69	68	70	77	87	90	89	91	89	87
27	87	87	85	81	79	81	93	95	95	95	91	85	82	79	76	76	80	87	92	93	94	94	94	95
28	94	94	94	93	92	93	93	93	94	82	75	71	63	82	85	84	89	91	91	92	92	88	86	80
29	79	83	85	90	94	95	95	96	95	94	93	92	93	88	86	83	81	86	89	90	92	93	93	93
30	93	90	90	86	83	83	84	84	74	82	70	65	65	60	63	63	60	75	82	86	84	82	84	84
31	82	87	78	80	68	64	63	54	48	42	40	38	36	34	33	34	35	37	33	33	35	37	35	38
1	35	31	30	31	30	30	37	43	44	49	69	71	69	70	71	73	74	83	83	80	79	80	86	86
2	93	94	94	94	93	91	86	86	82	75	66	69	63	63	79	68	74	83	90	91	91	92	92	92
3	93	93	93	93	93	92	90	90	88	81	64	57	52	65	58	60	60	58	68	76	75	71	72	75
4	80	77	78	76	75	70	70	68	54	41	37	33	31	29	26	41	46	55	64	66	70	70	61	57
5	62	60	57	59	59	62	69	61	57	45	53	61	55	50	53	55	65	74	84	87	83	76	76	78
6	80	80	80	78	77	82	85	84	77	68	62	63	60	59	64	72	80	84	87	85	83	86	85	86
7	87	76	80	76	74	68	67	63	64	38	28	25	25	62	53	45	60	62	65	59	57	47	40	42
8	46	52	44	34	49	61	49	41	43	40	36	49	45	42	41	35	36	42	38	53	48	56	64	64
9	60	59	59	51	55	56	55	44	51	46	38	44	39	35	61	54	52	67	66	61	72	65	72	64
10	56	49	44	47	39	46	39	35	35	29	24	24	22	19	21	21	30	30	32	38	40	35	35	35
11	36	35	35	32	30	27	26	28	26	23	19	13	32	49	66	74	80	83	79	80	82	85	90	90
12	92	79	79	85	87	90	87	89	83	79	69	69	64	68	67	69	77	84	78	90	91	93	92	90
13	92	94	94	94	93	92	90	87	85	75	71	65	64	65	66	66	67	74	82	85	88	87	89	90
14	90	91	90	90	89	91	91	91	86	73	61	66	68	71	71	69	72	80	83	87	85	88	87	84
15	85	87	85	86	85	87	87	88	84	73	71	68	65	62	61	65	64	73	76	71	75	81	81	80
16	79	78	79	79	80	77	79	79	65	65	57	62	57	59	66	68	38	48	59	62	69	73	72	73
17	73	77	75	79	70	74	74	75	63	51	62	60	57	62	59	61	65	59	65	66	73	76	76	76
18	77	75	79	76	81	79	80	80	67	68	63	59	60	58	60	61	73	71	71	75	76	78	76	77
19	72	76	78	77	78	82	81	73	57	48	46	41	38	46	58	57	60	66	71	70	63	67	67	63
20	60	59	57	59	56	55	56	56	46	42	38	34	24	19	16	13	25	31	33	41	44	42	45	35
21	50	58	69	78	73	90	94	94	93	88	79	76	73	80	78	81	85	87	90	91	91	93	94	94
22	94	95	95	95	95	96	93	91	92	91	91	88	90	86	88	88	84	81	84	84	76	73	75	78
23	80	83	88	89	87	88	90	92	93	93	94	94	94	95	94	93	86	88	92	93	94	94	94	94
24	94	94	94	94	95	94	94	94	93	85	74	66	65	67	66	69	73	75	84	88	91	92	92	91
25	93	92	93	93	93	94	94	94	92	89	83	79	77	74	76	72	73	78	86	89	89	91	93	95
26	95	96	96	96	96	96	96	96	95	94	90	79	73	70	76	72	71	75	80	81	86	89	89	90

Hours of the day – Nortier humidity data January 2002 – August 2002

Day	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*11*	*12*	*13*	*14*	*15*	*16*	*17*	*18*	*19*	*20*	*21*	*22*	*23*	*24*
27	89	89	91	89	87	86	85	86	77	67	68	67	65	65	63	69	74	80	87	88	91	92	92	92
28	93	93	94	94	92	92	92	90	84	70	74	68	66	68	70	70	79	84	86	87	91	93	94	94
29	94	91	87	85	80	79	76	75	66	57	57	58	61	59	63	67	71	75	79	89	93	94	94	94
30	94	94	94	93	92	90	92	89	80	72	82	65	61	50	67	73	62	70	74	84	82	86	84	84
31	83	84	81	82	84	85	85	81	70	58	52	41	55	52	51	60	64	72	81	84	87	85	86	81

Table B. 2: Nortier humidity data

C APPENDIX : AMMONIA REFRIGERATION

C.1 Ammonia Refrigeration Pipes

The outside temperature of ammonia pipes was measured and natural convection inside the plant was assumed.

Plant conditions:

Dry-bulb temperature	Wet-bulb temperature
$T_{dbp} := 288.15\,K$	$T_{nbp} := 286.55\,K$

Blue pipe at blast freezers:

The blue pipe carries the refrigerant, which is less than –10 °C and is evaporated in the blast freezers. Only calculations for the blue pipe were done to see the effect of natural convection inside the plant on heat gain, which results in inefficiencies. Temperatures of 0 °C, 9 °C and 11 °C respectively, were measured on the insulation on the outside of surfaces. In order to determine the heat loss caused by natural convection, one needs to determine the areas of the pipes at certain temperatures. To ease calculations, the pipe is divided in three segments of different diameters.

Segment 1:

	Surface area:
$d_1 := 0.45\,m$	$A_1 := \pi \cdot d_1 \cdot l_1$
$l_1 := 15\,m$	$A_1 = 21.206\,m^2$

Table C. 1: Dimensional properties of segment 1

The percentage of area at 0 °C and 9 °C is 15% and 85% respectively. The size of each area at 0 °C and 9 °C can be calculated as follows:

$A_{10} := 0.15 \cdot A_1$	$A_{19} := 0.85 \cdot A_1$
$A_{10} = 3.181\,m^2$	$A_{19} = 18.025\,m^2$

Table C. 2: Sizes of isothermal areas in segment 1

Segment 2:

	Surface area:
$d_2 := 0.25\,m$	$A_2 := \pi \cdot d_2 \cdot l_2$

$l_2 := 11\ \mathrm{m}$	$A_2 = 8.639\,\mathrm{m}^2$

Table C. 3: Dimensional properties of segment 2

The percentage of area at 0 °C and 9 °C is 20% and 80% respectively. The size of each area at 0 °C and 9 °C can be calculated as follows:

$A_{20} := 0.20 \cdot A_2$	$A_{29} := 0.8 \cdot A_2$
$A_{20} = 1.728\,\mathrm{m}^2$	$A_{29} = 6.912\,\mathrm{m}^2$

Table C. 4: Sizes of isothermal areas in segment 2

Segment 3:

	Surface area:
$d_3 := 0.25\ \mathrm{m}$	$A_3 := \pi \cdot d_3 \cdot l_3$
$l_3 := 11\ \mathrm{m}$	$A_3 = 8.639\,\mathrm{m}^2$

Table C. 5: Dimensional properties of segment 3

The percentage of area at 0 °C and 11 °C is 5% and 95% respectively. The size of each area at 0 °C and 9 °C can be calculated:

$A_{30} := 0.05 \cdot A_3$	$A_{311} := 0.95 \cdot A_3$
$A_{30} = 0.432\,\mathrm{m}^2$	$A_{311} = 8.207\,\mathrm{m}^2$

Table C. 6: Sizes of isothermal areas in segment 3

The total area of the segments at 0 °C, 9 °C and 11 °C can be seen below in Table C.7:

$A_0 := A_{10} + A_{20} + A_{30}$	$A_0 = 5.341\,\mathrm{m}^2$
$A_9 := A_{19} + A_{29}$	$A_9 = 24.936\,\mathrm{m}^2$
$A_{11} := A_{311}$	$A_{11} = 8.207\,\mathrm{m}^2$

Table C. 7: Area summation for isothermal areas

Natural Convection over Horizontal Cylinder

Assuming ideal gas conditions for the outside air, for an ideal gas, the volumetric coefficient of expansion is:

$$\beta := \frac{1}{T_{dbp}} \tag{C.1}$$

Thermo-physical properties of air at given temperatures are as follows:

Conduction coefficient	$k := 0.026 \frac{W}{m \cdot K}$
Kinematic viscosity	$\nu := 14.7 \cdot 10^{-6} \frac{m^2}{s}$
Prandtl number	$Pr := 0.69$
Gravitational acceleration	$g = 9.807 \frac{m}{s^2}$

Table C. 8: Thermal properties of air

Natural convection for segments 1, 2 and 3:

$\Delta T_0 := T_{dbp} - 273.15K$	$\Delta T_0 = 15\ K$
$\Delta T_9 := T_{dbp} - 282.15K$	$\Delta T_9 = 6\ K$
$\Delta T_{11} := T_{dbp} - 284.15K$	$\Delta T_{11} = 4\ K$

Table C. 9: Temperature differences for isothermal areas

Grashof, Rayleigh and Nusselt numbers for segment 1 at 0 °C:

$Gr_{D0} := \frac{\left(\beta \cdot \Delta T_0 \cdot g \cdot d_1^{\,3}\right)}{\nu^2}$	$Gr_{D0} = 2.153 \cdot 10^8$
$Ra_{D0} := Gr_{D0} \cdot Pr$	$Ra_{D0} = 1.485 \cdot 10^8$
$Nu_{D0} := 0.36 + \frac{\left(0.518 \cdot Ra_{D0}^{\;0.25}\right)}{\left[1 + \left(\frac{0.559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}}$	$Nu_{D0} = 43.471$
$h_{c0} := k \cdot \frac{Nu_{D0}}{d_1}$	$h_{c0} = 2.512 \frac{kg}{s^3 \cdot K}$

Table C. 10: Convection heat transfer for segment 1 areas at 0 °C

Grashof, Rayleigh and Nusselt numbers for segment 1 at 9 °C:

$Gr_{D9} := \frac{\left(\beta \cdot \Delta T_{9} \cdot g \cdot d_{1}^{3}\right)}{\nu^{2}}$	$Gr_{D9} = 8.611 \cdot 10^{7}$
$Ra_{D9} := Gr_{D9} \cdot Pr$	$Ra_{D9} = 5.942 \cdot 10^{7}$
$Nu_{D9} := 0.36 + \frac{\left(0.518 \cdot Ra_{D9}^{0.25}\right)}{\left[1 + \left(\frac{0.559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}}$	$Nu_{D9} = 34.645$
$h_{c9} := k \cdot \frac{Nu_{D9}}{d_{1}}$	$h_{c9} = 2.002 \frac{kg}{s^{3} \cdot K}$

Table C. 11: Convection heat transfer for segment 1 areas at 9 °C

From the values obtained in tables C.10 and C.11, the heat gain for segment 1 is calculated:

$$Q_{1} := h_{c0} \cdot A_{10} \cdot \Delta T_{0} + h_{c9} \cdot A_{19} \cdot \Delta T_{9} \qquad \text{(C.2)}$$

with

$Q_{1} = 336.324 W$

Grashof, Rayleigh and Nusselt numbers for segment 2 at 0 °C:

$Gr_{D20} := \frac{\left(\beta \cdot \Delta T_{0} \cdot g \cdot d_{2}^{3}\right)}{\nu^{2}}$	$Gr_{D20} = 3.691 \cdot 10^{7}$
$Ra_{D20} := Gr_{D20} \cdot Pr$	$Ra_{D20} = 2.547 \cdot 10^{7}$
$Nu_{D20} := 0.36 + \frac{\left(0.518 \cdot Ra_{D20}^{0.25}\right)}{\left[1 + \left(\frac{0.559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}}$	$Nu_{D20} = 28.102$
$h_{c20} := k \cdot \frac{Nu_{D20}}{d_{2}}$	$h_{c20} = 2.923 \frac{kg}{s^{3} \cdot K}$

Table C. 12: Convection heat transfer for segment 2 areas at 0 °C

Grashof, Rayleigh and Nusselt numbers for segment 2 at 9 °C:

$Gr_{D29} := \frac{\left(\beta \cdot \Delta T_{9} \cdot g \cdot d_{2}^{3}\right)}{\nu^{2}}$	$Gr_{D29} = 1.477 \cdot 10^{7}$
$Ra_{D29} := Gr_{D29} \cdot Pr$	$Ra_{D29} = 1.019 \cdot 10^{7}$
$Nu_{D29} := 0.36 + \frac{\left(0.518 \cdot Ra_{D29}^{0.25}\right)}{\left[1 + \left(\frac{0.559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}}$	$Nu_{D29} = 22.422$
$h_{c29} := k \cdot \frac{Nu_{D29}}{d_{2}}$	$h_{c29} = 2.332 \frac{kg}{s^{3} \cdot K}$

Table C. 13: Convection heat transfer for segment 2 areas at 9 °C

From the values obtained in tables C.12 and C.13, the heat gain for segment 2 is calculated:

$$Q_{2} := h_{c20} \cdot A_{20} \cdot \Delta T_{0} + h_{c29} \cdot A_{29} \cdot \Delta T_{9} \qquad (C.3)$$

with

$Q_{2} = 172.451\,W$

Grashof, Rayleigh and Nusselt numbers for segment 3 at 0 °C:

$Gr_{D30} := \frac{\left(\beta \cdot \Delta T_{0} \cdot g \cdot d_{3}^{3}\right)}{\nu^{2}}$	$Gr_{D30} = 3.691 \cdot 10^{7}$
$Ra_{D30} := Gr_{D30} \cdot Pr$	$Ra_{D30} = 2.547 \cdot 10^{7}$
$Nu_{D30} := 0.36 + \frac{\left(0.518 \cdot Ra_{D30}^{0.25}\right)}{\left[1 + \left(\frac{0.559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}}$	$Nu_{D30} = 28.102$
$h_{c30} := k \cdot \frac{Nu_{D30}}{d_{3}}$	$h_{c30} = 2.923 \frac{kg}{s^{3} \cdot K}$

Table C. 14: Convection heat transfer for segment 3 areas at 0 °C

Grashof, Rayleigh and Nusselt numbers for segment 3 at 11 °C:

$Gr_{D311} := \frac{\left(\beta \cdot \Delta T_{11} \cdot g \cdot d_3^{\,3}\right)}{\nu^2}$	$Gr_{D311} = 9.843 \cdot 10^6$
$Ra_{D311} := Gr_{D311} \cdot Pr$	$Ra_{D311} = 6.792 \cdot 10^6$
$Nu_{D311} := 0.36 + \frac{\left(0.518 \cdot Ra_{D311}^{\;0.25}\right)}{\left[1 + \left(\frac{0.559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}}$	$Nu_{D311} = 20.296$
$h_{c311} := k \cdot \frac{Nu_{D311}}{d_3}$	$h_{c311} = 2.111 \frac{kg}{s^3 \cdot K}$

Table C. 15: Convection heat transfer for segment 3 areas at 11 ^oC

From the values obtained in tables C.14 and C.15, the heat gain for segment 3 is calculated:

$$Q_3 := h_{c30} \cdot A_{30} \cdot \Delta T_0 + h_{c311} \cdot A_{311} \cdot \Delta T_{11} \qquad (C.4)$$

with

$$Q_3 = 88.232\,W$$

The total heat gain for the pipe under the specified plant conditions is obtained by summating equations (C.2) to (C.4):

$$Q_{total} := Q_1 + Q_2 + Q_3 \qquad (C.5)$$

which gives:

$$Q_{total} = 597.007\,W$$

The working hours per annum, t_h, with an availability of 95% and 350 days a year, is calculated as follows:

$$t_h := \left(350\,\text{day} \cdot 24\,\frac{\text{hr}}{\text{day}}\right) \cdot 0.95 \qquad \text{(C.6)}$$

with

$$t_h = 2.873 \cdot 10^7 \text{ s}$$

Keep in mind that on the day of measurement, the plant's environment was colder than normal conditions, therefore the results are conservative. Energy loss per year:

$$E_{loss} := Q_{total} \cdot t_h \qquad \text{(C.7)}$$

$$E_{loss} = 1.715 \cdot 10^{10} \text{ J}$$

Energy costs are calculated per unit of energy used, with one unit equals one kW energy, therefore:

1 unit = 1 kW x hr = 3.6 x 10^6 J (C.8)

Therefore, the number of energy units per annum is obtained by dividing equation (C.7) by equation (C.8):

$$E_{units} := \frac{E_{loss}}{\text{kW} \cdot \text{hr}} \qquad \text{(C.9)}$$

$$E_{units} = 4.764 \cdot 10^3$$

To determine an average price per unit for the plant, an old account was used:

Units	Price per unit	Price				
244973	0.0809	19818.32				
93562	0.3649	34140.77				
241919	0.1394	33723.51				
580454	0.151059	87682.6	Average			

Table C. 16: Average unit price for plant's electricity use

The average price per unit as calculated above is R0.151059 and can be used to determine the possible energy savings when new insulation material is implemented. With new insulation, the total savings per annum would be:

$$Unit_{price} := 0.151059 \qquad (C.10)$$

Total savings per year:

$$S_{year} := E_{units} \cdot Unit_{price} \qquad (C.11)$$

$$S_{year} = 719.663$$

Thus, R720 could be saved. To determine the amount of capital, P, that can be spent for a payback period of two years with an annual payback of A = R720 and an interest rate of 14.5%:

$S_{year} = 719.663$	$i_{year} := 0.145$
$n := 2$	$P := S_{year} \cdot \left[\frac{\left[\left(1 + i_{year} \right)^n - 1 \right]}{i_{year} \cdot \left(1 + i_{year} \right)^n} \right]$
$P = 1.177 \cdot 10^3$	

Table C. 17: Payback period calculation

The amount of capital that can be spent for a payback period of two years is R1 177. Take a look at how much capital can be spent when assuming that insulation material will last for 10 years:

$S_{year} = 719.663$	$i_{year} := 0.145$
$n := 10$	$P := S_{year} \cdot \left[\frac{\left[\left(1 + i_{year} \right)^n - 1 \right]}{i_{year} \cdot \left(1 + i_{year} \right)^n} \right]$
$P = 3.682 \cdot 10^3$	

Table C. 18: MathCAD calculations to calculate capital

For 10 years' payback R3 682 can be spent. The savings would be too little for any company to consider new insulation material, but from a good housekeeping point of view it is worthwhile to consider replacing the material as it degrades.

D Appendix: Calculations For Unlagged Steam Piping

D.1 Calculations for Unlagged Steam Piping

The heat loss of each uniform section of unlagged steam pipe will be determined, either as natural convection over a horizontal cylinder or as natural convection over a vertical wall. Firstly, the fixed geometrical properties will be calculated. For segment i, diameter d_i and length l_i are measured to determine the exposed area, A_i. The dry and wet-bulb temperatures, $T_{dbplant}$ and $T_{nbplant}$ respectively, are measured with a dry and wet-bulb sling thermometer.

Plant environment:

$T_{dbplant} := 288.15\,K$	$T_{nbplant} := 286.55\,K$
Exposed unit area	$A_i := 1\ m^2$
Pipe temperature	$T_i := 370\,K, 375\,K .. 480\,K$
Temperature difference	$\Delta T_i(T_i) := T_i - T_{dbplant}$
Pipe diameter	$d_i := 0.01\,m, 0.02\,m .. 0.2\,m$
Pipe length	$l_i := 1\ m$
Volumetric coefficient of expansion for ideal gas	$\beta := \frac{1}{T_{dbplant}}$
Kinematic viscosity for air	$\nu_{air} := 14.7 \cdot 10^{-6}\ \frac{m^2}{s}$
Thermal conductivity of air	$k_{air} := 0.026 \frac{W}{m \cdot K}$
Prandtl number	$Pr_{air} := 0.69$
Gravitational acceleration	$g = 9.807 \frac{m}{s^2}$

Table D. 1: Dimensional and thermal properties of pipe and plant atmosphere

D.1.1 Natural Convection over Horizontal Cylinder

In this case, the pipe diameter and pipe temperature are variables, which will determine laminar flow or turbulent flow. For the Rayleigh number smaller than 10^9, laminar flow exists, and for $Ra_d > 10^9$, transition inclines towards turbulent flow.

Exposed area: $$A_{di}(d_i) := \pi \cdot (2 \cdot d_i) \cdot l_i \qquad \text{(D.1)}$$

Grashof number: $$Gr_d(T_i, d_i) := \frac{\left(\beta \cdot \Delta T_i(T_i) \cdot g \cdot d_i^3\right)}{\nu_{air}^2} \qquad \text{(D.2)}$$

Rayleigh number: $$Ra_d(T_i, d_i) := Gr_d(T_i, d_i) \cdot Pr_{air} \qquad \text{(D.3)}$$

Let us assume that either laminar flow or turbulent flow occurs, but not both. Then:

Laminar Nusselt number: $$Nu_{dlam}(T_i, d_i) := 0.36 + \frac{\left(0.518 \cdot Ra_d(T_i, d_i)^{\frac{1}{4}}\right)}{\left[1 + \left(\frac{0.559}{Pr_{air}}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}} \qquad \text{(D.4)}$$

Turbulent Nusselt number: $$Nu_{dtur}(T_i, d_i) := \left[0.60 + 0.387 \cdot \left[\frac{Ra_d(T_i, d_i)}{\left[\left[1 + \left(\frac{0.559}{Pr_{air}}\right)^{\frac{9}{16}}\right]\right]^{\frac{16}{9}}}\right]^{\frac{1}{6}}\right]^2 \qquad \text{(D.5)}$$

$$Nu_d(T_i, d_i) := \begin{cases} Nu_{dlam}(T_i, d_i) & \text{if } 10^{-6} < Ra_d(T_i, d_i) \le 10^9 \\ Nu_{dtur}(T_i, d_i) & \text{if } Ra_d(T_i, d_i) > 10^9 \end{cases} \qquad \text{(D.6)}$$

With the Nusselt number known, the heat transfer coefficient can be calculated:

$$h_{cd}(T_i, d_i) := \left(\frac{k_{air}}{d_i}\right) \cdot Nu_d(T_i, d_i) \qquad \text{(D.7)}$$

Then, the heat loss is calculated as follows:

$$Q_{dloss}(T_i, d_i) := h_{cd}(T_i, d_i) \cdot A_{di}(d_i) \cdot \Delta T_i(T_i) \qquad \text{(D.8)}$$

D.1.2 Natural Convection over Vertical Wall

For a vertical cylinder with natural convection, one can approximate the cylinder as a vertical wall:

Exposed area: $$A_{Li}(d_i) := \pi \cdot (2 \cdot d_i) \cdot l_i \qquad \text{(D.9)}$$

Grashof number:

$x := 0.1\ \text{m}$ $$Gr_L(T_i) := \frac{\left(\beta \cdot \Delta T_i(T_i) \cdot g \cdot l_i^{\,3}\right)}{\nu_{air}^{\;2}} \qquad \text{(D.10)}$$

Rayleigh number: $$Ra_L(T_i) := Gr_L(T_i) \cdot Pr_{air} \qquad \text{(D.11)}$$

Transition from laminar to turbulent flow will occur at Gr_X~10^9 with the Prandtl number function:

$$\Psi := \left[1 + \left(\frac{0.492}{Pr_{air}}\right)^{\frac{9}{16}}\right]^{\frac{-16}{9}} \qquad \text{(D.12)}$$

with

$\Psi = 0.343$

Laminar Nusselt number: $$Nu_{Llam}(T_i) := 0.68 + 0.67 \cdot \left(Ra_L(T_i) \cdot \Psi\right)^{\frac{1}{4}} \qquad \text{(D.13)}$$

Turbulent Nusselt number: $$Nu_{Ltur}(T_i) := 0.68 + 0.67 \cdot \left(Ra_L(T_i) \cdot \Psi\right)^{\frac{1}{4}} \cdot \left(1 + 1.6 \cdot 10^{-8} \cdot Ra_L(T_i) \cdot \Psi\right)^{\frac{1}{12}}$$

(D.14)

The Nusselt number is:

$$Nu_L(T_i) := \begin{cases} Nu_{Llam}(T_i) & \text{if } Ra_L(T_i) \le 10^9 \\ Nu_{Ltur}(T_i) & \text{if } 10^9 < Ra_L(T_i) < 10^{12} \end{cases} \tag{D.15}$$

with the heat transfer coefficient reflected by:

$$h_{cL}(T_i) := \left(\frac{k_{air}}{l_i}\right) \cdot Nu_L(T_i) \tag{D.16}$$

and the heat loss to the environment:

$$Q_{Lloss}(T_i, d_i) := h_{cL}(T_i) \cdot A_{Li}(d_i) \cdot \Delta T_i(T_i) \tag{D.17}$$

For a typical pipe of 50 mm diameter and with 8 bar steam pressure, or an equivalent of 443.5 K (170.2 °C), the energy loss for 1 m of unlagged piping is:

$$Q_{dloss}(443.5\,\mathrm{K}, 0.05\,\mathrm{m}) = 386.864\,\mathrm{W} \tag{D.18}$$

$$Q_{Lloss}(443.5\,\mathrm{K}, 0.05\,\mathrm{m}) = 343.09\,\mathrm{W} \tag{D.19}$$

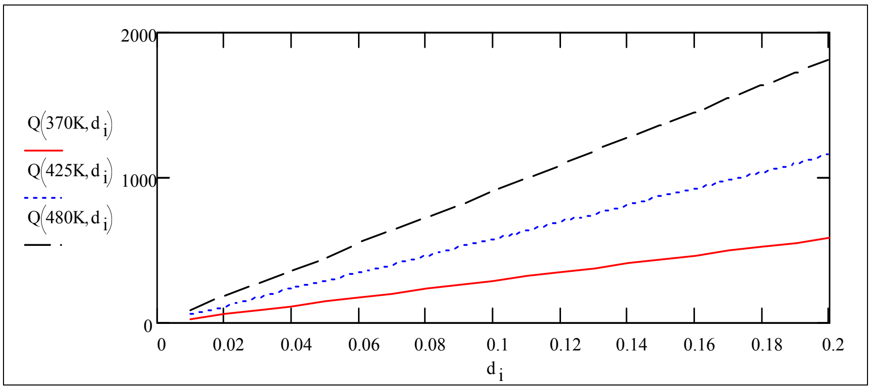


Figure D. 1: Convection heat loss: variable pipe diameter

From figure D.1 it is clear that convection heat loss is directly proportional to pipe diameter at constant temperatures.

D.1.3 Effect of Radiation on Energy Losses for Unlagged Steam Pipes

The stainless-steel piping will be subject to solar and sky irradiation. The solar radiation absorptance for stainless steel (Mills, 1995) and the hemispherical emmitance and reflectivity are given in the table below:

.

Absorptivity for stainless steel	$\alpha_{ss} := 0.89$
Hemispherical emittance	$\varepsilon_s := 0.20$
Reflectivity	$\rho_s := 1 - \alpha_{ss}$
Stefan-Boltzmann constant	$\sigma := 5.670 \cdot 10^{-8} \dfrac{W}{m^2 \cdot K^4}$

Table D. 2: Radiation properties of stainless steel (Mills, 1995)

Calculating the Emmissivity of the Sky

For a dry-bulb temperature of 288.15 K (15 ^oC) and a wet-bulb temperature of 286.55 K (13.4 ^oC), the relative humidity is 85% from the psychrometric chart, and the saturated vapour pressure at 288.15 K is 1704.4 Pa. Therefore, the partial pressure of water vapour is:

$$P_{vapour} := 0.85 \cdot 1704.4 Pa \tag{D.20}$$

with atmospheric pressure:

$$P_{atm} := 101300 Pa \tag{D.21}$$

According to Mills, 1995, the sky emissivity is:

$$\varepsilon_{sky} := 0.55 + 1.8 \cdot \left(\frac{P_{vapour}}{P_{atm}} \right)^{\frac{1}{2}} \tag{D.22}$$

$$\varepsilon_{sky} = 0.765$$

Therefore, sky irradiation is:

$$G_{sky} := \varepsilon_{sky} \cdot \sigma \cdot T_{dbplant}^{\;4} \qquad \text{(D.23)}$$

The total radiation heat flux is represented by the following:

$$q_{radpipe}\left(T_i\right) := \varepsilon_s \cdot \sigma \cdot T_i^{\;4} + \rho_s \cdot G_{sky} - \alpha_{ss} \cdot G_{sky} \qquad \text{(D.24)}$$

With total radiation heat transfer:

$$Q_{rad}\left(T_i, d_i\right) := q_{radpipe}\left(T_i\right) \cdot A_{di}\left(d_i\right) \qquad \text{(D.25)}$$

Radiative heat transfer per 1 m length of steam pipe at 0.8 MPa (8 bar) absolute pressure and with a 50 mm outside pipe diameter:

$$Q_{rad}(443.5\,K, 0.05\,m) = 64.527\,W$$

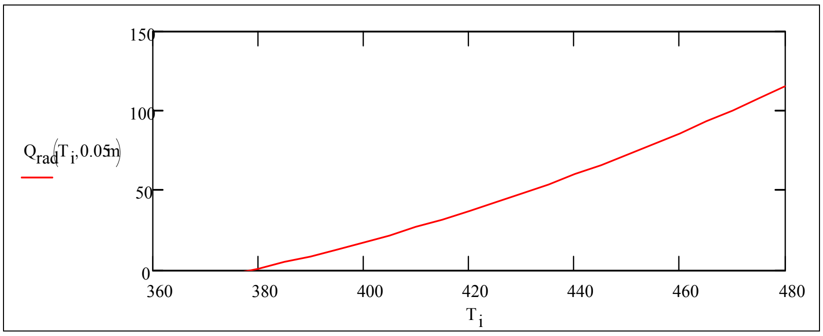


Figure D. 2: Radiation heat loss: variable temperature

From figure D.2, it is evident that an exponential relationship exists between radiation heat loss and surface temperature.

Total Heat Transfer – Radiative and Convective – for Steam Pipe of 1 m Length

Because most steam pipes are placed horizontally, it is only necessary to look at horizontal steam pipes.

$$Q_{total}(T_i, d_i) := Q_{dloss}(T_i, d_i) + Q_{rad}(T_i, d_i) \qquad \text{(D.26)}$$

For a horizontal steam pipe at 8 bar steam pressure and with a diameter of 50 mm, the heat loss per meter is:

$$Q_{total}(443.5\,K, 0.05\,m) = 451.391\,W \qquad \text{(D.27)}$$

Cost of Steam Generated from Coal

Consider a coal price of R447 per ton. The calorific value of A-grade pea coal is 28.5 MJ/kg. For 1 W of steam saved at a utilisation point, ignoring inefficiencies (except boiler efficiency), the amount of coal that is used will be:

Boiler efficiency: $\eta_{boiler} := 0.85$ (D.28)

Calorific value of pea coal: $Cal_{pea} := 28.5 \cdot 10^6 \frac{J}{kg}$ (D.29)

$Usage_{rate} := \frac{(1\,W)}{Cal_{pea} \cdot \eta_{boiler}}$ $Usage_{rate} = 4.128 \cdot 10^{-8} \frac{kg}{s}$ (D.30)

$Coal_{price} := \frac{0.447}{kg}$ (D.31)

$Cost_{rate} := Usage_{rate} \cdot Coal_{price}$ $Cost_{rate} = 1.845 \cdot 10^{-8}\,Hz$ (D.32)

Cost of 1 m of Unlagged Piping

Once again, a typical pipe will be used to indicate losses. The cost will be evaluated on a period of one year:

$$Yearly_{cost} := 300\,day \cdot \frac{Cost_{rate}}{W} \cdot Q_{total}(443.5\,K, 0.05\,m) \qquad \text{(D.33)}$$

$$Yearly_{cost} = 215.89 \qquad \text{(D.34)}$$

Thus, for 1 m of steam pipe (8 bar, 0.05 m diameter) that is exposed to the plant environment, it will cost a plant an additional R215.89 per annum for energy consumption.

Percentage Analysis of Heat Transfer Losses from Unlagged Steam Pipes

Comparisons of radiative heat transfer to convective heat transfer, as well as between vertical and horizontal piping, are indicated below.

Horizontal-to-vertical ratio:

$$\frac{Q_{dloss}(443.5\,K, 0.05\,m)}{Q_{Lloss}(443.5\,K, 0.05\,m)} = 1.128 \tag{D.35}$$

Radiative-to-convective heat transfer ratio for horizontal piping:

$$\frac{Q_{rad}(443.15\,K, 0.05\,m)}{Q_{rad}(443.5\,K, 0.05\,m) + Q_{dloss}(443.5\,K, 0.05\,m)} = 0.142 \tag{D.36}$$

Conclusions

From the above calculations, the following conclusions can be drawn:

1. The heat loss for vertical and horizontal piping is almost the same, although horizontal piping has more convective heat loss to the environment.
2. For an unlagged steam pipe of 1 m and with an outside diameter of 50 mm and 0.8 MPa (8 bar) absolute saturated steam pressure, assuming the temperature on the outside is almost the steam temperature, radiative heat loss is ± 14% of the total heat loss of 451.391 W and the cost of the heat loss is equivalent to R215.89 per annum.

E Appendix: Heat Load Calculations for Insulation Panels

E.1 Calculation of Heat Load on Outside Panels of Cooling Room

Heat on the outside panel is transferred through radiation and convection. By measuring the dry and wet-bulb temperature and assuming correlations for irradiation on the panel during different times of the day, recognising the direction to which the perpendicular on the panel points, it is possible to estimate the amount of heat that travels through the insulation into the cooling room.

Geometry of insulation panel:

$L_{wall} := 7\ m$	$W_{wall} := 7\ m$
$A_{wall} := L_{wall} \cdot W_{wall}$	

Table E. 1: Dimensional properties of insulation panel

Temperature measurements and atmospheric conditions:

$T_{dbo} := 287.15 K$	$Twbo := 286.75 K$
$P_{atm} := 101325 Pa$	$u_{wind} := 0.2 \frac{m}{s}$
$T_{po} := 284.75 K$	$T_{dbi} := 253.15 K$
$RH_i := 0.56$	$T_{pi} := 255.15 K$

Table E. 2: Temperature measurements and atmospheric conditions

Radiation properties:

$\varepsilon_p := 0.85$	$\alpha_p := 0.25$

Table E. 3: Radiation properties of insulation panel

Assume an average relative humidity of 83%:

$$RH := 0.83 \qquad (E.1)$$

Correlation for determining saturated vapour pressure in the cooling room:

$C_{1} := -5.674535910^{3}$	$C_{2} := -5.152305810^{-1}$
$C_{3} := -9.677843010^{-3}$	$C_{4} := 6.221570110^{-7}$
$C_{5} := 2.074782510^{-9}$	$C_{6} := -9.484024010^{-13}$
$C_{7} := 4.1635019$	$C_{8} := -5.800220610^{3}$
$C_{9} := -5.51625610^{0}$	$C_{10} := -4.864023910^{-2}$
$C_{11} := 4.176476810^{-5}$	$C_{12} := -1.445209310^{-8}$
$C_{13} := 6.5459673$	

Table E. 4: Coefficients for calculation of saturated vapour pressure

$$P_{vsat}(T) := 1000 \cdot \left[e^{\left(\frac{C_{8} \cdot K}{T}\right) + C_{9} + C_{10} \cdot \frac{T}{K} + C_{11} \cdot \frac{T^{2}}{K^{2}} + C_{12} \cdot \frac{T^{3}}{K^{3}} + C_{13} \cdot \ln\left(\frac{T}{K}\right)} \cdot Pa \right] \qquad \text{(E.2)}$$

Using perfect gas relations for the dry and moist air,

$$P_{vsat}\left(T_{dbo}\right) = 1.599 \cdot 10^{3}\ Pa \qquad \text{(E.3)}$$

from equations E.1 and E.2, the vapour pressure is obtained:

$$P_{v} := RH \cdot P_{vsat}\left(T_{dbo}\right) \qquad \text{(E.4)}$$

$$P_{v} = 1.327 \cdot 10^{3}\ Pa$$

From a correlation in Mills, 1995, the sky emissivity is:

$$\varepsilon_{sky} := 0.55 + 1.8 \cdot \left(\frac{P_{v}}{P_{atm}}\right) \qquad \text{(E.5)}$$

$$\varepsilon_{sky} = 0.574$$

Properties of air:

density:

$$\rho_{air}(T) := \frac{P_{atm}}{287.08 \frac{J}{kg \cdot K} \cdot T} \qquad \text{(E.6)}$$

$$\rho_{air}(T_{dbo}) = 1.229 \frac{kg}{m^3}$$

dynamic viscosity:

$$\mu_{air}(T) := \left[2.287973 \cdot 10^{-6} + 6.259793 \cdot 10^{-8} \cdot \frac{T}{K} - 3.131956 \cdot 10^{-11} \cdot \left(\frac{T}{K}\right)^2 + 8.15038 \cdot 10^{-15} \cdot \left(\frac{T}{K}\right)^3\right] \frac{kg}{s \cdot m}$$

(E.7)

kinematic viscosity:

$$\nu_{air}(T) := \frac{\mu_{air}(T)}{\rho_{air}(T)} \qquad \text{(E.8)}$$

thermal conductivity:

$$k_{air}(T) := \left[-4.937787 \cdot 10^{-4} + 1.018087 \cdot 10^{-4} \cdot \frac{T}{K} - 4.627937 \cdot 10^{-8} \cdot \left(\frac{T}{K}\right)^2 + 1.250603 \cdot 10^{-11} \cdot \left(\frac{T}{K}\right)^3\right] \frac{W}{m \cdot K}$$

(E.9)

specific heat:

$$c_{pair}(T) := \left[1.045356 \cdot 10^{3} - 3.161783 \cdot 10^{-1} \cdot \frac{T}{K} + 7.083814 \cdot 10^{-4} \cdot \left(\frac{T}{K}\right)^2 - 2.705209 \cdot 10^{-7} \cdot \left(\frac{T}{K}\right)^3\right] \frac{J}{kg \cdot K}$$

(E.10)

Prandtl number: $$\Pr(T) := \frac{\left(c_{pair}(T) \cdot \mu_{air}(T)\right)}{k_{air}(T)} \qquad \text{(E.11)}$$

Prandtl number function: $$\Psi(T) := \left[1 + \left(\frac{0.492}{\Pr(T)}\right)^{\frac{9}{16}}\right]^{\frac{-16}{9}} \qquad \text{(E.12)}$$

Volumetric expansion coefficient: $$\beta := \frac{-\left(\rho_{air}(T_{pi}) - \rho_{air}(T_{dbo})\right)}{\rho_{air}(T_{pi}) \cdot (T_{pi} - T_{dbo})} \qquad \text{(E.13)}$$

$$\beta = 3.483 \cdot 10^{-3} \frac{1}{K}$$

Grashof number:

$x := 1\ m$

$$Gr_x(x) := \frac{\left[\beta \cdot (T_{pi} - T_{dbo}) \cdot g \cdot x^3\right]}{\nu_{air}(T_{dbo})^2} \qquad \text{(E.14)}$$

$$Gr_L := Gr_x(L_{wall}) \tag{E.15}$$

Rayleigh number: $$Ra_L(T_{dbo}) := Gr_L \cdot Pr(T_{dbo}) \tag{E.16}$$

Reynolds number: $$Re_x(x) := \frac{(u_{wind} \cdot x)}{\nu_{air}(T_{dbo})} \tag{E.17}$$

There are two scenarios for convection heat transfer: natural convection and forced convection, with an assumed average wind speed. Calculations are done for both.

Natural Convection (Laminar and Turbulent Flow)

Point of transition for natural convection:

$$\mathrm{root}\left(Gr_x(x) - 10^9, x\right) = -0.578\,\mathrm{m}$$

$$x_{tr} := \mathrm{root}\left(Gr_x(x) - 10^9, x\right)$$

Laminar Nusselt number:

$$Nu_{Lnatlam} := 0.68 + 0.670 \cdot \left(Ra_L(T_{dbo}) \cdot \Psi(T_{dbo})\right)^{\frac{1}{4}} \tag{E.18}$$

Turbulent Nusselt number:

$$Nu_{Lnattur} := 0.68 + 0.670 \cdot \left(Ra_L(T_{dbo}) \cdot \Psi(T_{dbo})\right)^{\frac{1}{4}} \cdot \left(1 + 1.6 \cdot 10^{-8} \cdot Ra_L(T_{dbo}) \cdot \Psi(T_{dbo})\right)^{\frac{1}{12}} \tag{E.19}$$

$$Nu_{Lnatural} := \begin{vmatrix} Nu_{Lnatlam} \ \text{ if } \ Ra_L(T_{dbo}) \le 10^9 \\ Nu_{Lnattur} \ \text{ if } \ 10^9 < Ra_L(T_{dbo}) \le 10^{12} \end{vmatrix} \tag{E.20}$$

Because the wall is 7 m high, for the sake of simplicity, we ignore the laminar part and assume turbulent flow for the full height of the wall.

Forced Convection (Laminar and Turbulent Flow)

For plate flow, there will always be a laminar and turbulent region due to the length of the plate. The point of transition can be solved with the root function.

$$Re_{trans} := 50000 \tag{E.21}$$

$$x_{trturb} := \mathrm{root}\left(Re_x(x) - Re_{trans}, x\right) \qquad x_{trturb} = 3.635\,\mathrm{m} \tag{E.22}$$

Average Nusselt number:

$$Nu_{Lforced} := 0.664\,Re_x\left(x_{trturb}\right)^{\frac{1}{2}} \cdot Pr\left(T_{dbo}\right)^{\frac{1}{3}} + 0.036\,Re_x\left(L_{wall}\right)^{0.8} \cdot Pr\left(T_{dbo}\right)^{0.43} \cdot \left[1 - \left(\frac{Re_x\left(x_{trturb}\right)}{Re_x\left(L_{wall}\right)}\right)^{0.8}\right] \tag{E.23}$$

Nusselt number:

$$Nu_L := \begin{vmatrix} Nu_{Lnatural} & \text{if } u_{wind} = 0\,\frac{m}{s} \\ Nu_{Lforced} & \text{if } u_{wind} > 0\,\frac{m}{s} \end{vmatrix} \tag{E.24}$$

Convection heat transfer coefficient:

$$h_{conv} := \frac{\left(k_{air}\left(T_{dbo}\right) \cdot Nu_L\right)}{L_{wall}} \tag{E.25}$$

$$h_{conv} = 0.922\,\frac{kg}{s^3 \cdot K}$$

Heat transfer due to convection:

$$q_{conv} := h_{conv} \cdot \left(T_{dbo} - T_{po}\right) \tag{E.26}$$

$$q_{conv} = 2.213\,\frac{kg}{s^3}$$

Radiation Heat Transfer

Radiation heat transfer comprises solar irradiation, sky irradiation and emission from the surface. The solar irradiation, G_S, is a function of the time of day and year.

$$G_s := 19 \frac{W}{m^2} \qquad \text{(E.27)}$$

$$q_{rad} := \begin{vmatrix} \left(\left(\alpha_p \cdot \varepsilon_{sky} \cdot \sigma \cdot T_{dbo}{}^4 + \alpha_p \cdot G_s - \varepsilon_p \cdot \sigma \cdot T_{po}{}^4\right)\right) & \text{if } T_{po} > T_{dbo} \\ \left(\alpha_p \cdot \varepsilon_{sky} \cdot \sigma \cdot T_{dbo}{}^4 + \alpha_p \cdot G_s\right) & \text{if } T_{po} \le T_{dbo} \end{vmatrix}$$

(E.28)

$$q_{rad} = 60.024 \frac{kg}{s^3}$$

Overall Heat Gain from Atmosphere Next to Panel

$$q_{total} := q_{conv} + q_{rad} \qquad \text{(E.29)}$$

$$q_{total} = 62.237 \frac{kg}{s^3}$$

With the overall heat gain calculated, it is possible to determine the overall heat transfer coefficient, U_{actual}, by dividing q_{total} by the temperature difference,

T_{po} - T_{pi}:

$$U_{actual} := \frac{q_{total}}{T_{po} - T_{pi}} \qquad \text{(E.30)}$$

$$U_{actual} = 2.103 \frac{kg}{s^3 \cdot K}$$

Calculation of Design Value for Insulation

The insulation consists of polystyrene foam with steel plates on both sides, coated with white paint. The steel plates are glued to the polystyrene foam, but for the sake of simplicity, we will ignore the effect of the paint and the glue on the heat transfer coefficient of the insulation panel.

Thickness of steel plates	$t_s := 0.0015\,m$
Thickness of polystyrene	$t_p := 0.150\,m$
Thermal conductivity of steel	$k_s := 15 \frac{W}{m \cdot K}$
Thermal conductivity of polystyrene	$k_p := 0.04 \frac{W}{m \cdot K}$

Table E. 5: Properties of insulation panel

The design value of the overall heat transfer coefficient is as follows:

$$U_{design} := \left[\frac{t_p}{k_p} + \frac{(2 \cdot t_s)}{k_s}\right]^{-1} \qquad \text{(E.31)}$$

$$U_{design} = 0.267 \frac{kg}{s^3 \cdot K}$$

For the panels on the eastern side of cold stores 1 and 3, an extra layer of panels of 50 mm is added:

$$t_{pextra} := 200\,mm \qquad \text{(E.32)}$$

Replacing t_s in equation (E.31) with t_{pextra} in equation (E.32), the overall heat transfer coefficient for the southern wall is obtained:

$$U_{designsouth} := \left(\frac{4 \cdot t_s}{k_s} + \frac{t_{pextra}}{k_p}\right)^{-1} \qquad \text{(E.33)}$$

$$U_{designsouth} = 0.2 \frac{kg}{s^3 \cdot K}$$

Comparison of the design value and the actual values:

$$\text{Percentage} := \frac{U_{actual}}{U_{designsouth}} \tag{E.34}$$

$$\text{Percentage} = 10.514$$

This means that if the inside temperature is -18 ^{o}C and the outside temperature is the same as the environment temperature at night (14 ^{o}C), then:

$$Q_{total} := U_{designsouth} \cdot \left(T_{po} - T_{pi}\right) \tag{E.35}$$

$$Q_{total} = 5.92 \frac{kg}{s^3}$$

The percentage improvement, assuming that new insulation panels are performing at design values, is as follows:

$$\text{Improvement} := \frac{\left(q_{total} - Q_{total}\right) \cdot 100}{Q_{total}} \tag{E.36}$$

$$\text{Improvement} = 951.382$$

Thus, the insulation effect will be improved by 951% if new insulation is installed.

F Appendix: Cold Store Calculations

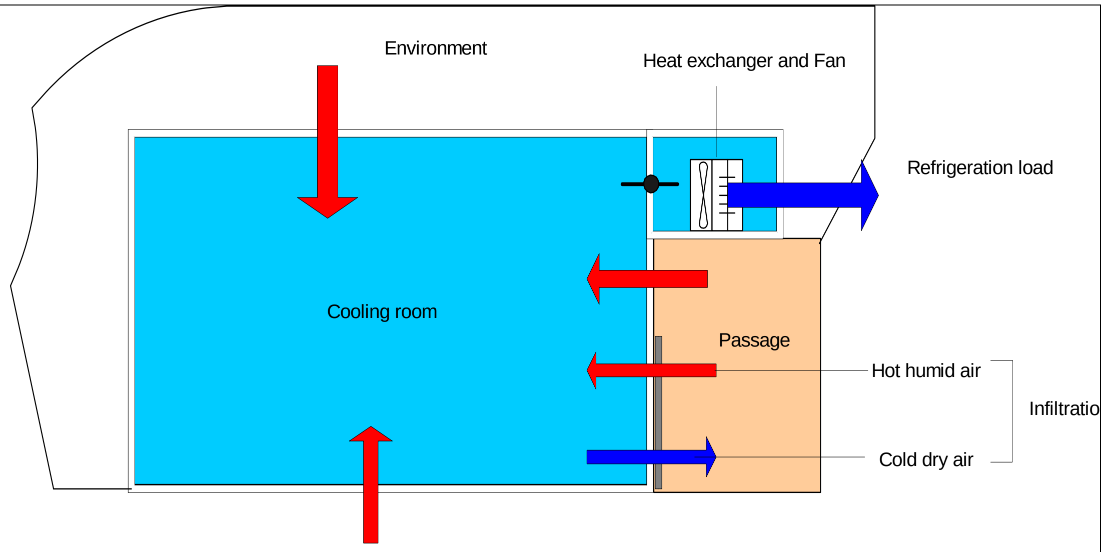
F.1 Cold Store Calculations

The purpose of the calculations is to determine the effect of water vapour on air that enters the cooling rooms through infiltration effects and which is condensed and frozen in the cold stores, as well as heat loss through poor insulation.

F.1.1 Infiltration Effect

1) The specific humidity of the passage and cooling room must be calculated.

2) The potential for mass transfer can now be used to determine infiltration.



- Important to determine the effect of poor insulation on the heat load of the refrigeration by measuring surface temperatures all around the cooling room and calculating the conduction coefficient of each side and determining the loss to the environment.
- Temperatures on the in and outsides must be compared with design specifications to determine cost savings possible.
- To determine the refrigeration load, one can measure the mass flow of the refrigerant in the heat exchanger and the inlet and outlet temperatures and pressures, or the air flow across the heat exchanger can be determined and the inlet and outlet wet and dry bulb temperatures can be measured.
- The question is whether or not all these abovementioned equipment is really neccessary. Is it possible to determine mass flowrate at the outlet of the compressor ?

Figure F. 1: Schematic representation of heat transfer of cold store

$T_{dbc} := 248.15K$	$T_{evap} := 236.15K$
$T_{dbp} := 287.113K$	$T_{nbp} := 286.152K$

$P_{atm} := 101325\,Pa$	

Table F. 1: Plant and cold store conditions

Correlation for determining vapour pressure in the cooling room:

$C_1 := -5.6745359\cdot10^{3}$	$C_2 := -5.1523058\cdot10^{-1}$
$C_3 := -9.6778430\cdot10^{-3}$	$C_4 := 6.2215701\cdot10^{-7}$
$C_5 := 2.0747825\cdot10^{-9}$	$C_6 := -9.4840240\cdot10^{-13}$
$C_7 := 4.1635019$	$C_8 := -5.8002206\cdot10^{3}$
$C_9 := -5.516256\cdot10^{0}$	$C_{10} := -4.8640239\cdot10^{-2}$
$C_{11} := 4.1764768\cdot10^{-5}$	$C_{12} := -1.4452093\cdot10^{-8}$
$C_{13} := 6.5459673$	

Table F. 2: Constants for saturated vapour pressure

The vapour pressure can be calculated from the following correlations (Kröger, 1998):

$$P_{vicesat}(T) := 1000\cdot\left[e^{\left(\frac{C_1\cdot K}{T}\right)+C_2+C_3\cdot\frac{T}{K}+C_4\cdot\frac{T^2}{K^2}+C_5\cdot\frac{T^3}{K^3}+C_6\cdot\frac{T^4}{K^4}+C_7\cdot\ln\left(\frac{T}{K}\right)}\cdot Pa\right] \tag{F.1}$$

$$P_{vliqsat}(T) := 1000\cdot\left[e^{\left(\frac{C_8\cdot K}{T}\right)+C_9+C_{10}\cdot\frac{T}{K}+C_{11}\cdot\frac{T^2}{K^2}+C_{12}\cdot\frac{T^3}{K^3}+C_{13}\cdot\ln\left(\frac{T}{K}\right)}\cdot Pa\right] \tag{F.2}$$

Using perfect gas relations for the dry and moist air,

$$P_{vsat}(T) := \begin{cases} P_{vicesat}(T) & \text{if } T \le 273.15K \\ P_{vliqsat}(T) & \text{if } T > 273.15K \end{cases} \tag{F.3}$$

cold store vapour pressure is obtained by applying equations F.1, F.2 and F.3 for T_{dbc}:

$$P_{vcsat} := P_{vsat}\left(T_{dbc}\right) \tag{F.4}$$

$$P_{vcsat} = 63.289\,Pa$$

The passage vapour pressure is obtained in the same way as above, but for T_{dbp}.

$$P_{vpsat} := P_{vsat}\left(T_{dbp}\right) \tag{F.5}$$

$$P_{vpsat} = 1.595 \cdot 10^{3}\ Pa$$

With the saturation vapour pressure known for the cooling room and the passage, the saturated specific humidity can be found respectively

$$\omega_{csat} := \frac{0.62198 P_{vcsat}}{P_{atm} - P_{vcsat}} \tag{F.6}$$

$$\omega_{csat} = 3.887 \cdot 10^{-4}$$

$$\omega_{psat} := \frac{0.62198 P_{vpsat}}{P_{atm} - P_{vpsat}} \tag{F.7}$$

$$\omega_{psat} = 9.946 \cdot 10^{-3}$$

The specific heat capacity for dry air:

$$c_{pa} := 1006 \frac{J}{kg \cdot K} \tag{F.8}$$

Calculation of Relative Humidity in Cold Store

Assuming that the dew-point temperature of the cold store approximates the evaporator surface temperature, T_{evap}, the vapour pressure in the cold store can be approximated and, therefore, the wet-bulb temperature can be calculated. Constants from ASHRAE Fundamentals (1993), used to calculate the dew-point temperature of liquid and ice:

$C_{14} := 6.54$	$C_{15} := 14.526$
$C_{16} := 0.7389$	$C_{17} := 0.09486$
$C_{18} := 0.4569$	

Table F. 3: Constants to determine dew-point temperature

A first value is selected as P_{vcroot} from where MathCAD starts an iterative process to determine the true vapour pressure P_{vc}, setting the dew-point temperature equal to the evaporator temperature:

$$P_{vcroot} := P_{vcsat} \cdot 0.5 \tag{F.9}$$

$$T_{dice}\left(P_{vcroot}\right) := \left[6.09 + 12.608 \ln\left(\frac{P_{vcroot}}{1000\,Pa}\right) + 0.4959 \left(\ln\left(\frac{P_{vcroot}}{1000\,Pa}\right)\right)^2 + 273.15\right] K \tag{F.10}$$

$$T_{dliq}\left(P_{vcroot}\right) := \left[C_{14} + C_{15} \cdot \ln\left(\frac{P_{vcroot}}{1000\,Pa}\right) + C_{16} \cdot \left(\ln\left(\frac{P_{vcroot}}{1000\,Pa}\right)\right)^2 + C_{17} \cdot \left(\ln\left(\frac{P_{vcroot}}{1000\,Pa}\right)\right)^3 + C_{18} \cdot \left(\frac{P_{vcroot}}{1000\,Pa}\right)^{0.1984} + 273.15\right] K \tag{F.11}$$

$$T_{dew}\left(P_{vcroot}\right) := \begin{cases} T_{dice}\left(P_{vcroot}\right) & \text{if } T_{dbc} < 273.15K \\ T_{dliq}\left(P_{vcroot}\right) & \text{if } T_{dbc} \geq 273.15K \end{cases} \tag{F.12}$$

$$\mathrm{root}\left(T_{dew}\left(P_{vcroot}\right) - T_{evap}, P_{vcroot}\right) = 17.097\,Pa \tag{F.13}$$

$$P_{vc} := \mathrm{root}\left(T_{dew}\left(P_{vcroot}\right) - T_{evap}, P_{vcroot}\right) \tag{F.14}$$

thus,

$$RH_c := \frac{P_{vc}}{P_{vcsat}} \tag{F.15}$$

$$RH_c = 0.27$$

Determining the specific humidity:

$$\omega_c := \frac{(0.62198 P_{vc})}{P_{atm} - P_{vc}} \qquad \text{(F.16)}$$

$$\omega_c = 1.05 \cdot 10^{-4}$$

The conservation of energy in adiabatic saturation of air at a known dry-bulb temperature and relative humidity is applied. Specific enthalpy for dry air:

$$h_a(T) := c_{pa} \cdot (T - 273.15\,K) \qquad \text{(F.17)}$$

The specific enthalpy for water vapour is demonstrated by:

$$h_v(T) := \left[2501 \cdot 10^3 + 1805 \cdot \frac{(T - 273.15\,K)}{K}\right] \frac{J}{kg} \qquad \text{(F.18)}$$

The specific enthalpy for ice and water liquid respectively:

$$h_{fice}(T) := \left(1888.27 \cdot \frac{T}{K} - 849.88 \cdot 10^3\right) \frac{J}{kg} \qquad \text{(F.19)}$$

$$h_{fliq}(T) := \left(4183.9 \cdot \frac{T}{K} - 1142.7 \cdot 10^3\right) \frac{J}{kg} \qquad \text{(F.20)}$$

For temperatures above 273.15 K equation F.20 is used, otherwise F.19 is used:

$$h_f(T) := \begin{vmatrix} h_{fice}(T) & \text{if } T \le 273.15\,K \\ h_{fliq}(T) & \text{if } T > 273.15\,K \end{vmatrix} \qquad \text{(F.21)}$$

The specific enthalpy for evaporation is shown below:

$$h_{fg}(T) := \begin{cases} 1000\left[-0.0038\left(\frac{T}{K}\right)^2 + 1.8164\frac{T}{K} + 2621.9\right]\frac{J}{kg} & \text{if } T \le 273.15K \\ \left[\left[1000\left(3151.5 - 2.3814\frac{T}{K}\right)\frac{J}{kg}\right]\right] & \text{if } T > 273.15K \end{cases} \quad \text{(F.22)}$$

The specific enthalpy values for dry air and water vapour are as follows:

$h_{a1} := h_a(T_{dbc})$	$h_{a1} = -2.515 \cdot 10^4$ Sv
$h_{v1} := h_v(T_{dbc})$	$h_{v1} = 2.456 \cdot 10^6$ Sv

Table F. 4: Specific enthalpy values for dry air and water vapour

Calculation of Wet-bulb Temperature in Cold Store

For calculation purposes in MathCAD, a value for the wet-bulb temperature was guessed, which was substituted into equation (F.23) and changed to find the roots of the equation, in order to find the actual wet-bulb temperature.

$T_{nbcroot} := 200\,K$	$\omega_{csat}(T_{nbcroot}) := \frac{0.62198 P_{vsat}(T_{nbcroot})}{P_{atm} - P_{vsat}(T_{nbcroot})}$
$h_{a2}(T_{nbcroot}) := h_a(T_{nbcroot})$	$h_{v2}(T_{nbcroot}) := h_v(T_{nbcroot})$
$h_{f2}(T_{nbcroot}) := h_f(T_{nbcroot})$	$h_{fg2}(T_{nbcroot}) := h_{fg}(T_{nbcroot})$

Table F. 5: Equations to determine wet-bulb temperature

$$f_c(T_{nbcroot}) := \left(\omega_c - \omega_{csat}(T_{nbcroot})\right) \cdot h_{f2}(T_{nbcroot}) + h_{a2}(T_{nbcroot}) + \omega_{csat}(T_{nbcroot}) \cdot h_{v2}(T_{nbcroot}) - h_{a1} - \omega_c \cdot h_{v1} \quad \text{(F.23)}$$

$$\text{root}\left(f_c(T_{nbcroot}), T_{nbcroot}\right) = 247.427K \quad \text{(F.24)}$$

From equation (F.24) we have the wet-bulb temperature, and equation (F.25) represents the wet-bulb temperature of the cold store:

$$T_{nbc} := \text{root}\left(f_c(T_{nbcroot}), T_{nbcroot}\right) \quad \text{(F.25)}$$

$$T_{nbc} = 247.427K$$

Calculation of Specific Humidity of Passage Air

$h_{a1} := h_a(T_{dbp})$ $h_{a2} := h_a(T_{nbp})$	$h_{a1} = 1.405 \cdot 10^4$ Sv $h_{a2} = 1.308 \cdot 10^4$ Sv
$h_{v1} := h_v(T_{dbp})$	$h_{v1} = 2.526 \cdot 10^6$ Sv
$h_{v2} := h_v(T_{nbp})$	$h_{v2} = 2.524 \cdot 10^6$ Sv
$h_{f2} := h_f(T_{nbp})$	$h_{f2} = 5.453 \cdot 10^4$ Sv
$h_{fg2} := h_{fg}(T_{nbp})$	
$\omega_{pnsat} := \dfrac{0.62198 P_{vsat}(T_{nbp})}{P_{atm} - P_{vsat}(T_{nbp})}$	$\omega_{pnsat} = 9.333 \cdot 10^{-3}$
$\omega_p := \dfrac{\left[h_{a2} - h_{a1} + \omega_{pnsat} \cdot (h_{fg2})\right]}{h_{v1} - h_{f2}}$	$\omega_p = 8.936 \cdot 10^{-3}$
$RH_p := \dfrac{\omega_p}{\omega_{psat}}$	$RH_p = 0.898$

Table F. 6: Calculation of specific and relative humidity of passage air

Partial Pressure of Water Vapour

The partial pressure of the water vapour in the cold store is:

$$P_{vc} := \frac{(P_{atm} \cdot \omega_c)}{0.62198 + \omega_c} \tag{F.26}$$

$P_{vc} = 17.097$ Pa

and the partial pressure of water vapour in the passage is:

$$P_{vp} := \frac{(P_{atm} \cdot \omega_p)}{0.62198 + \omega_p} \tag{F.27}$$

$P_{vp} = 1.435 \cdot 10^3$ Pa

The enormous difference between the vapour pressure of the cold store and that of the passage is the driving force behind vapour transfer into the cold store.

F.1.2 Mass Transfer

With the specific humidities of the cold store and the passage known, the mass transfer of water vapour from the passage to the cold store can be determined. Approach the problem as the diffusion of one gas through a second stagnant gas (ASHRAE Fundamentals Handbook, 1993).

Fick's law for ideal gases with negligible pressure gradient states the following:

$$J_B := -\rho \cdot D_v \cdot \frac{d}{dy}\left(\frac{\rho_B}{\rho}\right) \qquad \text{(F.28)}$$

$$J_{BC} := -C \cdot D_v \cdot \frac{d}{dy}\left(\frac{C_B}{C}\right) \qquad \text{(F.29)}$$

This is the basic equation for molecular diffusion, expressing the concentration of component B of a binary mixture in terms of the mass fraction (ρ_B/ρ) or mole fraction (C_B/C). D_v, the proportionality factor, is the mass diffusivity or the diffusion coefficient. The diffusive mass flux and diffusive molar flux are:

$J_B \equiv \rho_B \cdot (v_B - v)$ and $J_{BC} \equiv C_B \cdot (v_B - v)$

where $(v_B - v)$ is the velocity of component B relative to the mixture velocity. Remember that $C_B \equiv \frac{\rho_B}{M_B}$, and thus all results can be converted to the molar form and otherwise. Let's look at the physical situation we have here:

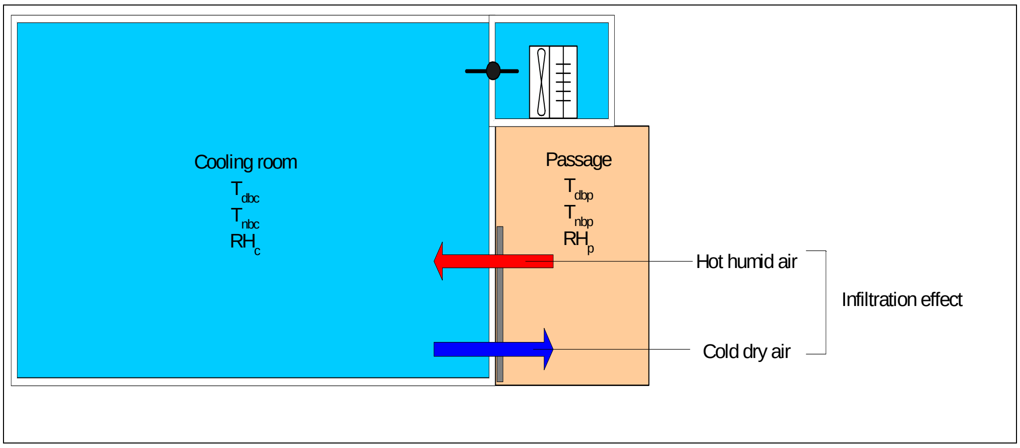


Figure F. 2: Schematic drawing of vapour transfer phenomena

The difference between concentration of water vapour in the cold store, C_{BCool}, and concentration of water vapour in the passage, C_{BPass}, will result in a net mass flux of vapour into the cold store, resulting in additional cooling load requirements. Fick's law on the cold store and the passage states:

$$J_{Bc} := -\rho_c \cdot D_{vc} \cdot \frac{d}{dy}\left(\frac{\rho_{Bc}}{\rho_c}\right) \tag{F.30}$$

$$J_{Bp} := -\rho_p \cdot D_{vp} \cdot \frac{d}{dy}\left(\frac{\rho_{Bp}}{\rho_p}\right) \tag{F.31}$$

from

$$\omega := \frac{m_B}{m_a} \tag{F.32}$$

and

$$m_t := m_B + m_a \tag{F.33}$$

also, then $\rho := \rho_B + \rho_a$ and in a fixed volume we have $\frac{\rho_B}{\rho} = \frac{m_B}{m_t}$

and deviding above and under by m_a, we have $\frac{m_B}{m_t} = \frac{\omega}{\omega + 1}$.

Therefore, the mass flux can be calculated in terms of specific humidity.

$$J_{Bnet} := -\rho_c \cdot D_{vc} \cdot \frac{d}{dy}\left(\frac{\rho_{Bc}}{\rho_c}\right) + -\rho_p \cdot D_{vp} \cdot \frac{d}{dy}\left(\frac{\rho_{Bp}}{\rho_p}\right) \tag{F.34}$$

$$J_{Bnet} := -\rho_c \cdot D_{vc} \cdot \frac{d}{dy}\left(\frac{\omega_c}{\omega_c + 1}\right) + -\rho_p \cdot D_{vp} \cdot \frac{d}{dy}\left(\frac{\omega_p}{\omega_p + 1}\right) \tag{F.35}$$

The different densities of the cold store and the passage can be determined with the known dry-bulb and wet-bulb temperatures, where the calculation of the diffusion coefficient will follow.

Calculation of the Diffusion Coefficient

An empirical equation for the mass diffusivity of water vapour in air (ASHRAE Fundamentals Handbook, 1993) is as follows:

$$D_v := \left(\frac{0.926}{p}\right) \cdot \left(\frac{T^{2.5}}{T + 245}\right) \tag{F.36}$$

Density calculations for moist air in the cold store and passage provided the following results:

$$\rho_c := \left(1 + \omega_c\right) \cdot \left(1 - \frac{\omega_c}{\omega_c + 0.62198}\right) \cdot \frac{P_{atm}}{287.08 \frac{J}{kg \cdot K} \cdot T_{dbc}} \tag{F.37}$$

$$\rho_c = 1.422 \frac{kg}{m^3}$$

$$\rho_p := (1+\omega_p)\cdot\left(1-\frac{\omega_p}{\omega_p+0.62198}\right)\cdot\frac{P_{atm}}{287.08\frac{J}{kg\cdot K}\cdot T_{dbp}} \qquad \text{(F.38)}$$

$$\rho_p = 1.223\frac{kg}{m^3}$$

The diffusion coefficient for the air in the cold store and passage is given by:

$$D_{vc} := \left(\frac{0.926}{1000 \cdot P_{atm}}\right)\cdot\frac{\left(\frac{T_{dbc}^{2.5}}{T_{dbc}+245\,K}\right)\cdot kg\cdot m}{K^{1.5}\cdot s^3} \qquad \text{(F.39)}$$

$$D_{vc} = 1.798\cdot 10^{-5}\,\frac{m^2}{s}$$

$$D_{vp} := \left(\frac{0.926}{1000 \cdot P_{atm}}\right)\cdot\frac{\left(\frac{T_{dbp}^{2.5}}{T_{dbp}+245\,K}\right)\cdot kg\cdot m}{K^{1.5}\cdot s^3} \qquad \text{(F.40)}$$

$$D_{vp} = 2.399\cdot 10^{-5}\,\frac{m^2}{s}$$

Approximating d_y as very small, for example $d_y := 0.001\,m$, the net mass flux J_{Bnet} can be calculated:

$$J_{Bnet} := -\rho_c\cdot D_{vc}\cdot\frac{\left(\frac{\omega_c}{\omega_c+1}\right)}{d_y}+\rho_p\cdot D_{vp}\cdot\frac{\left(\frac{\omega_p}{\omega_p+1}\right)}{d_y} \qquad \text{(F.41)}$$

$$J_{Bnet} = 2.571\cdot 10^{-4}\,\frac{kg}{m^2\cdot s}$$

Therefore, the daily mass transfer of water vapour from the passage through the door to the cold store, with the assumption that the door is effectively open for three hours each day, is:

$$m_{transfer} := J_{Bnet}\cdot A_{door}\cdot t_{open} \qquad \text{(F.42)}$$

$$m_{transfer} = 20.16\,kg$$

$w_{door} := 2.2\,m$	$h_{door} := 3.3\,m$
$t_{open} := 3\,hr$	
$A_{door} := w_{door} \cdot h_{door}$	$A_{door} = 7.26\,m^2$

Table F. 7: Dimensional properties of door

Measurements for these specific values showed that the real mass transfer for each day is 210.6 kg, which is higher, because the convection caused by the forklifts was ignored in the calculations. Assume a linear correction factor to be applied:

$$Correc_{factor} := \frac{(210.6\,kg)}{m_{transfer}} \qquad (F.43)$$

$$Correc_{factor} = 10.446$$

For reasons of simplicity, this correction factor will be used to compensate for convection mass transfer.

In order to determine the average mass transfer rate, average temperature and humidity values for a year have been used.

$T_{dbc} := 248.15\,K$	$T_{evap} := 236.15\,K$
$T_{dbp} := 289.547\,K$	$T_{nbp} := 287.08\,K$
$P_{atm} := 101325\,Pa$	

Table F. 8: Average temperatures for a year

The values in table F.8 were obtained through the correlations between Nortier weather station and the plant atmosphere, and taking the average of the correlated plant values over the extended period. The same approach as above was followed to determine the average diffusive mass transfer for the year, and through applying the correction factor, the mass transfer for the year was approximated.

From table F.2 and equations (F.1) to (F.5), the average cold store and passage vapour pressure for a year was obtained.

The cold store vapour pressure is:

$$P_{vcsat} := P_{vsat}\left(T_{dbc}\right) \tag{F.44}$$

$$P_{vcsat} = 63.289\,\mathrm{Pa}$$

The passage vapour pressure is:

$$P_{vpsat} := P_{vsat}\left(T_{dbp}\right) \tag{F.45}$$

$$P_{vpsat} = 1.865 \cdot 10^{3}\ \mathrm{Pa}$$

With the saturation vapour pressure known for cold store and the passage, the saturated specific humidity can be found:

$$\omega_{csat} := \frac{0.62198 P_{vcsat}}{P_{atm} - P_{vcsat}} \tag{F.46}$$

$$\omega_{csat} = 3.887 \cdot 10^{-4}$$

$$\omega_{psat} := \frac{0.62198 P_{vpsat}}{P_{atm} - P_{vpsat}} \tag{F.47}$$

$$\omega_{psat} = 0.012$$

From equation (F.15), the relative humidity and the specific humidity of the cold store is known.

Conservation of energy in adiabatic saturation of air at a known dry-bulb temperature and relative humidity was applied by using equations (F.17) to (F.22).

The cold store wet-bulb temperature remains the same, and therefore the enthalpies remain the same.

Calculation of Specific Humidity of Passage Air

$h_{a1} := h_a(T_{dbp})$	$h_{a1} = 1.65 \cdot 10^4\ \text{Sv}$
$h_{a2} := h_a(T_{nbp})$	$h_{a2} = 1.401 \cdot 10^4\ \text{Sv}$
$h_{v1} := h_v(T_{dbp})$	$h_{v1} = 2.531 \cdot 10^6\ \text{Sv}$
$h_{v2} := h_v(T_{nbp})$	$h_{v2} = 2.526 \cdot 10^6\ \text{Sv}$
$h_{f2} := h_f(T_{nbp})$	$h_{f2} = 5.841 \cdot 10^4\ \text{Sv}$
$h_{fg2} := h_{fg}(T_{nbp})$	
$\omega_{pnsat} := \dfrac{0.62198 P_{vsat}(T_{nbp})}{P_{atm} - P_{vsat}(T_{nbp})}$	$\omega_{pnsat} = 9.925 \cdot 10^{-3}$
$\omega_p := \dfrac{\left[h_{a2} - h_{a1} + \omega_{pnsat} \cdot (h_{fg2})\right]}{h_{v1} - h_{f2}}$	$\omega_p = 8.903 \cdot 10^{-3}$
$RH_p := \dfrac{\omega_p}{\omega_{psat}}$	$RH_p = 0.763$

Table F. 9: Calculation of specific and relative humidity of passage air

Partial pressure of water vapour for the cold store and passage respectively:

$$P_{vc} := \frac{(P_{atm} \cdot \omega_c)}{0.62198 + \omega_c} \tag{F.48}$$

$P_{vc} = 17.097\,\text{Pa}$

$$P_{vp} := \frac{(P_{atm} \cdot \omega_p)}{0.62198 + \omega_p} \tag{F.49}$$

$P_{vp} = 1.43 \cdot 10^3\ \text{Pa}$

Mass Transfer

Thus, the mass flux can be calculated in terms of specific humidity as before.

$$J_{Bnet} := -\rho_c \cdot D_{vc} \cdot \frac{d}{dy}\left(\frac{\omega_c}{\omega_c + 1}\right) + -\rho_p \cdot D_{vp} \cdot \frac{d}{dy}\left(\frac{\omega_p}{\omega_p + 1}\right)$$

The different densities of the cooling room and the passage can be determined with the known dry-bulb and wet-bulb temperatures, upon which the calculation of the diffusion coefficient will follow. An empirical equation for the mass diffusivity of water vapour in air (ASHRAE, 1993) is as follows:

$$D_v := \left(\frac{0.926}{p}\right) \cdot \left(\frac{T^{2.5}}{T + 245}\right)$$

Density calculations for moist air in passage:

$$\rho_p := \left(1 + \omega_p\right) \cdot \left(1 - \frac{\omega_p}{\omega_p + 0.62198}\right) \cdot \frac{P_{atm}}{287.08 \frac{J}{kg \cdot K} \cdot T_{dbp}} \tag{F.50}$$

$$\rho_p = 1.212 \frac{kg}{m^3}$$

Diffusion coefficient for passage:

$$D_{vp} := \left(\frac{0.926}{1000 \cdot P_{atm}}\right) \cdot \frac{\left(\frac{T_{dbp}^{\ 2.5}}{T_{dbp} + 245\,K}\right) \cdot kg \cdot m}{K^{1.5} \cdot s^3} \tag{F.51}$$

$$D_{vp} = 2.439 \cdot 10^{-5} \frac{m^2}{s}$$

The net mass flux from equation (F.41) is:

$$J_{Bnet} = 2.583 \cdot 10^{-4} \frac{kg}{m^2 \cdot s} \tag{F.52}$$

Thus, the daily mass transfer of water vapour from the passage through the door to the cold store, with the assumption that the door is effectively open for three hours each day, is, for average values:

$$m_{transfer} = 20.252\,kg \tag{F.53}$$

Therefore, the average vapour transfer rate for infiltration alone is 20.252 kg.

Energy transfer due to different states of water vapour:

Water vapour gas constant $R_v := 461.5 \frac{J}{kg \cdot K}$ (F.54)

Critical pressure $P_{cr} := 22.09 \cdot 10^6\ Pa$ (F.55)

Critical temperature $T_{cr} := 647.3\,K$ (F.56)

Specific volume of vapour in cold store:

$$\upsilon_c := \frac{\left(R_v \cdot T_{dbc}\right)}{P_{vc}} \qquad \upsilon_c = 6.698 \cdot 10^3\ \frac{m^3}{kg} \tag{F.57}$$

Specific volume of vapour in passage:

$$\upsilon_p := \frac{\left(R_v \cdot T_{dbp}\right)}{P_{vp}} \qquad \upsilon_p = 92.326 \frac{m^3}{kg} \tag{F.58}$$

The internal energy change of water vapour is:

$$\Delta u_v := \left[a \cdot \left(\frac{T_{dbp}}{K} - \frac{T_{dbc}}{K} \right) + \left(\frac{b}{2} \right) \cdot \left[\left(\frac{T_{dbp}}{K} \right)^2 - \left(\frac{T_{dbc}}{K} \right)^2 \right] + a \cdot \left(\frac{-1}{\upsilon_c \cdot \frac{kg}{m^3}} + \frac{1}{\upsilon_p \cdot \frac{kg}{m^3}} \right) \right] \frac{J}{kg} \tag{F.59}$$

$$\Delta u_v = 6.644 \cdot 10^4\ Sv$$

where

$$a := \frac{\left(27 \cdot R_v{}^2 \cdot T_{cr}{}^2\right) \cdot kg \cdot s^2}{64 \cdot P_{cr} \cdot m^5} \qquad a = 1.704 \cdot 10^3 \tag{F.60}$$

and

$$b := \frac{\left(R_v \cdot T_{cr}\right) \cdot kg}{8 \cdot P_{cr} \cdot m^3} \qquad b = 1.69 \cdot 10^{-3} \tag{F.61}$$

The specific enthalpy change of water vapour is given by:

$$\Delta h_v := \Delta u_v + \left(P_{vp} \cdot \upsilon_p - P_{vc} \cdot \upsilon_c\right) \tag{F.62}$$

$$\Delta h_v = 8.442 \cdot 10^4 \ \mathrm{Sv}$$

The energy needed to freeze 1 kg of water vapour is:

$$E_{freeze} := 2.061 \cdot 10^6 \ \frac{J}{kg} \tag{F.63}$$

Enthalpy Increase in Cold Store

The enthalpy increase is given by:

$$E_{freeze} + \Delta h_v = 2.145 \cdot 10^6 \ \mathrm{Sv} \tag{F.64}$$

The mass of water that is condensed and frozen per day is as follows:

$$m_{peryear} := 400 \ kg \tag{F.65}$$

The average enthalpy increase per day is:

$$H_{experimental} := m_{peryear} \cdot \left(\Delta h_v + E_{freeze}\right) \tag{F.66}$$

The average enthalpy increase for one year is:

$$H_{experimentalperyear} := H_{experimental} \cdot \frac{365}{kW \cdot hr} \qquad (F.67)$$

$$H_{experimentalperyear} = 8.701 \cdot 10^4$$

In Rand-value, the energy cost just for removing moisture from the air is approximately R0,15:

$$Unit := 0.15 \qquad (F.68)$$

$$Cost := Unit \cdot H_{experimentalperyear} \qquad (F.69)$$

$$Cost = 1.305 \cdot 10^4$$

Thus, the extra energy cost due to water vapour infiltration amounts to R13 050 per year.

G Appendix: Payback Calculations for Air Curtains for Cold Stores

G.1 Payback Calculations of Air Curtains for the Cold Stores

Calculations are done to establish the amount that can be saved annually in terms of energy costs when the vapour transfer into the cold store is reduced by 80%. Maintenance costs are approximated as 10% of the capital costs for the first two years and thereafter 20%. Each unit uses 2.2 kW electrical energy to drive the fan motor and costs R4 950. The average unit price, assuming the operating time of the air curtains are spread evenly throughout the day, is 19 c. The average amount of water vapour condensed onto the evaporators is approximated at 211.55 kg per day, thus:

$$m_{freeze} := 211.52\,kg \qquad (G.1)$$

With an effectiveness of 80% in limiting vapour transfer to the cold store, the mass of water vapour kept from infiltrating the cold store is:

$$\eta_{eff} := 0.8 \qquad (G.2)$$

$$m_{saved} := \eta_{eff} \cdot m_{freeze} \qquad (G.3)$$

$$m_{saved} = 169.216\,kg$$

Assume the plant is operative for 350 days of the year, then the water that would have been condensed amounts to: 169.216 kg x 350 = 59234.67 kg. The energy value to condense vapour and freeze it to the temperature of the evaporator, T_{evap}, is:

$$e_{freeze} := 2947000\,\frac{J}{kg} \qquad (G.4)$$

The energy saved for one year is then:

$$E_{year} := e_{freeze} \cdot m_{saved} \cdot 350 \qquad (G.5)$$

$$E_{year} = 1.745 \cdot 10^{11}\,J$$

The amount of kWh saved for the year is as follows:

$$\text{Amount}_{\text{kWh}} := \frac{E_{\text{year}}}{\text{kW}\cdot\text{hr}} \qquad \text{(G.6)}$$

$$\text{Amount}_{\text{kWh}} = 4.848\cdot 10^{4}$$

Assume the average cost for 1 unit (1 kWh) is 19 c, then the infiltration cost will be:

$$\text{Cost}_{\text{infiltration}} := 0.19\cdot\text{Amount}_{\text{kWh}} \qquad \text{(G.7)}$$

$$\text{Cost}_{\text{infiltration}} = 9.212\cdot 10^{3}$$

For two air curtains, the total capital cost is:

$$\text{Cost}_{\text{aircurtains}} := 2\cdot 4950 \qquad \text{(G.8)}$$

The maintenance cost for two air curtains is:

$$\text{Cost}_{\text{maintenance}} := 0.1\cdot\text{Cost}_{\text{aircurtains}} \qquad \text{(G.9)}$$

Assume that the air curtains operate full-day, then the annual running cost of the air curtains will be:

$$\text{Cost}_{\text{electricity}} := \frac{(0.17\cdot 2.2\ \text{kW}\cdot 24\ \text{hr}\cdot 300)}{\text{kW}\cdot\text{hr}} \qquad \text{(G.10)}$$

The total running cost for each year for the first two years is the sum of equations (G.7) to (G.10), thus:

$$\text{Cost}_{\text{running}} := \text{Cost}_{\text{maintenance}} + \text{Cost}_{\text{electricity}} \qquad \text{(G.11)}$$

$$\text{Cost}_{\text{running}} = 3.683\cdot 10^{3}$$

The potential savings are determined by subtracting equation (G.11) from equation (G.7):

$$Cost_{savings} := Cost_{infiltration} - Cost_{running} \tag{G.12}$$

$$Cost_{savings} = 5.529 \cdot 10^{3}$$

A simple payback calculation gives:

$$n_{year} := \frac{Cost_{aircurtains}}{Cost_{savings}} \tag{G.13}$$

$$n_{year} = 1.791$$

Thus, in almost two years' time the air curtains would have paid for themselves. In addition, the safety factor of visibility, which as always has a financial risk attached to it, should be taken into consideration.

H APPENDIX: INVESTIGATION OF IMPROVEMENT OF STEAM UTILISATION

H.1 Investigation of Improvement of Steam Utilisation

The steam peeler uses steam to peel potatoes and then discards the steam into the atmosphere. This investigation will look into the possibility of utilising the steam for the blanchers or hot water.

H.1.1 Steam Peeler

Steam consumption at a production rate of 1 000 kg potatoes per hour requires 100 kg saturated steam at 1.6 MPa (16 bar) absolute pressure. Thus, for a current production rate of 100 t (100 000 kg) French fries per day, with an availability of 97% and 22 hours of production, the incoming product mass flow rate (equation (H.4)) is calculated:

$$Prod_{rate} := 100 \frac{ton}{day} \qquad (H.1)$$

$$availability := 0.97 \qquad (H.2)$$

$$yield := 0.5 \qquad (H.3)$$

$$Prod_{raw} := Prod_{rate} \cdot 24 \frac{hr}{day} \cdot \frac{availability}{yield} \qquad (H.4)$$

$$Prod_{raw} = 2.037 \frac{kg}{s}$$

The amount of steam used is as follows:

$$steam_{used} := 0.1\ kg \cdot \frac{Prod_{raw}}{1\ kg} \qquad (H.5)$$

$$steam_{used} = 0.204 \frac{kg}{s}$$

To determine the energy content of the steam that is discarded, one can approximate it with the design calculations of a Dutch company, Uticon-Dynatherm Industrie bv. By knowing the mass loss of the potatoes in the steam peeler, the steam that is used can be approximated. Due to the short time of exposure, it can be assumed that very little sensible heat is absorbed by the potatoes.

$$\text{Mass}_{\text{losspeeler}} := 0.14 \tag{H.6}$$

$$\text{steam}_{\text{discard}} := \left(1 - \text{Mass}_{\text{losspeeler}}\right) \cdot \text{steam}_{\text{used}} \tag{H.7}$$

$$\text{steam}_{\text{discard}} = 0.175 \frac{\text{kg}}{\text{s}}$$

In the industry, a rule of thumb for the assumption for coal consumption to steam generation is 1 ton of coal to produce 8 tons of steam. Thus, coal consumption:

$$\text{Coal}_{\text{consumed}} := \frac{\text{steam}_{\text{discard}}}{8} \tag{H.8}$$

With an average price of R447/ton coal:

$$\text{Value}_{\text{coal}} := \frac{447\ \text{A}}{\text{ton}} \tag{H.9}$$

From the above, the value of coal that is discarded can be calculated:

$$\text{Value}_{\text{coaldiscard}} := \text{Coal}_{\text{consumed}} \cdot \text{Value}_{\text{coal}} \cdot \text{day} \cdot \text{availability} \tag{H.10}$$

$$\text{Value}_{\text{coaldiscard}} = 904.252\,\text{A}$$

Assuming the boiler is working for 300 days of the year, then:

$$\text{Value}_{\text{coaldiscardperyear}} := \text{Value}_{\text{coaldiscard}} \cdot 300 \tag{H.11}$$

$$\text{Value}_{\text{coaldiscardperyear}} = 2.713 \cdot 10^{5}\ \text{A}$$

The question is, however, how will you practically use this energy that is discarded into the environment? One can capture this heat by means of a heat exchanger and produce hot water for the blanchers, as well as the hot water system. What is the energy usage of the blanchers and hot water system?

Hot Water System

For an output of 2 ton/h of French fries, the hot water system uses approximately 150 kg/h of steam at 0.2 MPa (2 bar) guage pressure, with an energy value of 113.5 kW, and defrosting uses 50 kg/h of steam at 0.4 MPa (4 bar) guage pressure, with an energy value of 38 kW. Therefore, a total of 151.5 kW is needed. Let us assume that there is a linear correlation between steam usage and production. For an output of 4.29 ton/h (the current production rate), a total of 151.5 kW x 4.29/2 = 325.4 kW is thus needed.

The enthalpy of saturated steam at 16 bar is:

$$h_{16barsat} := 2793880 \frac{J}{kg} \tag{H.12}$$

The potential energy of steam that is discarded is:

$$E_{potential} := steam_{discard} \cdot h_{16barsat} \tag{H.13}$$

$$E_{potential} = 4.894 \cdot 10^5 \frac{kg \cdot m^2}{s^3}$$

Thus, 489.4 kW steam energy is available and with an effectiveness of 80%, 391 kW can be used. This leads to the following cost savings:

$Energy_{steamsave} := 0.8 \cdot E_{potential}$	$Calorific_{coal} := 27.32 \cdot 10^6 \frac{J}{kg}$
$Coal_{save} := \frac{Energy_{steamsave}}{Calorific_{coal}}$	$Coal_{save} = 0.014 \frac{kg}{s}$
$Coal_{saveyear} := Coal_{save} \cdot day \cdot availability \cdot 300$	$Coal_{saveyear} = 3.603 \cdot 10^5\ kg$
$Money_{year} := Coal_{saveyear} \cdot \frac{400}{ton}$	$Money_{year} = 1.589 \cdot 10^5$

Table H. 1: Steam cost savings calculations

If the waste steam that is discarded and 80% of the heat are recaptured and transferred to the hot water system and blanchers, R158 900 will be saved per annum.

I Appendix: Calculations for Installation of Optical Sorter in Production Line

I.1 Calculations for Installation of Optical Sorter in Production Line

Installing an optical sorter in the production line will lead to higher sorting effectiveness. At this stage, 10 workers are employed to manually sort out the defects.

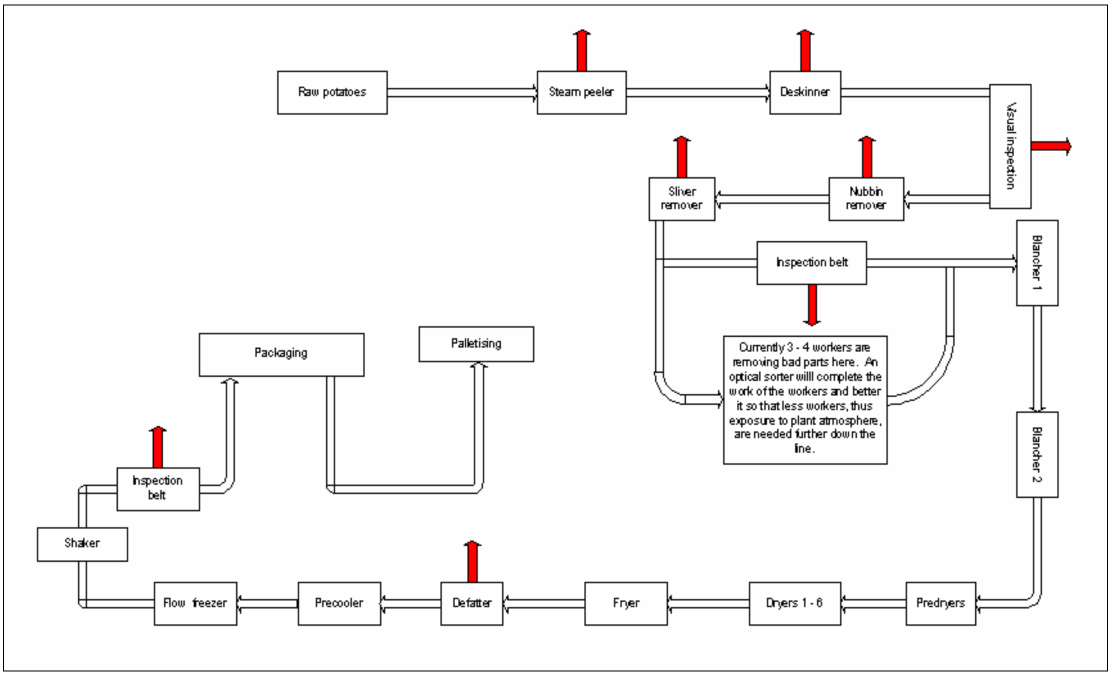


Figure I. 1: Production line with optical sorter inserted

Specifications for the fries are as follows:

2% of packed fries are defective.

Incoming defects through the year:

60% of year 15% defects, able to reach specification.

25% of year 20% defects, unable to reach specification, but can only get to 4% defects.

15% of year < 10% defects, able to reach specification.

Workers at the inspection line:

At inspection line 1, 6 workers are responsible for removing 60% of the defects.

At inspection line 2, 4 workers are responsible for removing 40% of the defects.

Temperature of fries from inspection belt 1:

After inspection belt 1:	
Blancher 1	20 ^oC
Blancher 2	
Blancher 3	70 ^oC, then drops to 55 ^oC
Predryers	(atmospheric exposure)
Dryers 1 – 6	40 ^oC (29.79% to evaporate of incoming product)
Fryer	100 ^oC (15.86% to evaporate of incoming product)
Defatter	
Precooler	
Flow freezer	-20 ^oC
Shaker	-14 ^oC
After inspection belt 2:	
Packaging	-6 ^oC
Palletising	-5 ^oC
Blast freezer	-20 ^oC
Cooling rooms	-20 ^oC

Table I. 1: Product temperatures on production line

Total percentage of defects taken as an average over a year:
60% per year at (15 - 2)% = 7.8%
25% per year at (20 - 4)% = 4%
15% per year at (5 - 2)% = 0.75%

Thus, the total defects over a year as average, η_{line}= 12.25 %.
Of this 12.25%, 60% (7.35% defects) is removed at inspection line 1, and 40% is removed at inspection line 2 (4.90% defects).

Defects Sorted with Optical Sorter

The optical sorter will remove all the defects at inspection line 1, whereas the 0.1% defects at inspection line 2 (formed by the fryer) will be removed manually, but now only one worker is needed, or that 0.1% can be left to fit into specifications. This means (12.25 - 0.1)% =

12.15% is removed at line 1. This difference in removed defects (12.15 - 7.35)% will cause an energy saving, as well as a saving of 9 workers' labour. The energy spent on 1 000 kg/h fries starting from inspection line are calculated as follow:

Energy Saved in Blancher

Specific heat of potato:

$$c_{ppotato} := 3780 \frac{J}{kg \cdot K} \tag{I.1}$$

Mass flow of potatoes in:

$$m_{pin} := 1000 \frac{kg}{hr} \tag{I.2}$$

Percentage of difference sorted out at inspection line:

$$\eta_{line1} := (0.1245 - 0.0753) \tag{I.3}$$

Product temperature difference in blanchers:

$$\Delta T_{blanchers} := (70 - 20)\ K \tag{I.4}$$

Mass flow of defects saved:

$$m_{saved} := \eta_{line1} \cdot m_{pin} \tag{I.5}$$

$$m_{saved} = 0.014 \frac{kg}{s}$$

Blancher energy saved:

$$E_{blancher} := m_{saved} \cdot c_{ppotato} \cdot \Delta T_{blanchers} \tag{I.6}$$

$$E_{blancher} = 2.583 \cdot 10^{3}\ W$$

Energy Saved in Dryer

Product temperature difference	$\Delta T_{dryers} := (40 - 55)\ K$
Percentage of mass of fries lost due to evaporation before dryer	$\eta_{evap} := (1 - 0.020833)$
Mass flow of product into dryer	$m_{dryerin} := \eta_{evap} \cdot m_{saved}$
Latent heat of evaporation of moisture	$h_{fgproduct} := 2260000 \frac{J}{kg}$
Percentage of mass lost in dryer	$\eta_{dryer} := (1 - 0.2979)$
Energy saved in dryer	
$E_{dryer} := \eta_{dryer} \cdot m_{dryerin} \cdot h_{fgproduct}$	$E_{dryer} = 2.123 \cdot 10^{4}\ W$

Table I. 2: Energy savings calculations for dryer

Energy Saved in Fryer

Mass flow into fryer	$m_{fryerin} := \eta_{dryer} \cdot m_{dryerin}$
Product temperature diference	$\Delta T_{fryer} := (100 - 40)\ K$
Percentage of mass evaporated	$\eta_{fryer} := (1 - 0.1586)$
Energy saved in fryer	
$E_{fryer} := \eta_{fryer} \cdot m_{fryerin} \cdot h_{fgproduct} + m_{fryerin} \cdot c_{ppotato} \cdot \Delta T_{fryer}$	$E_{fryer} = 2 \cdot 10^{4}\ W$

Table I. 3: Energy savings calculations for fryer

Energy Saved in Flow Freezer

Mass flow into freezer	$m_{flowfreezer} := \eta_{fryer} \cdot m_{fryerin}$
Product temperature difference	$\Delta T_{flowfreezer} := (40 - (-20))\ K$
Energy saved in flow freezer	
$E_{freezer} := m_{flowfreezer} \cdot c_{ppotato} \cdot \Delta T_{flowfreezer}$	$E_{freezer} = 1.793 \cdot 10^{3}\ W$

Table I. 4: Energy savings calculations for flow freezer

I.1.1 Conclusion

This is only the energy saved when looking at inspection line 1 and ignoring the gains in terms of shorter exposure of the final product, because fewer workers were needed to sort defects. The total energy saved for 1 000 kg/h fries just before inspection line 1:

$$E_{totalsaved} := E_{blancher} + E_{dryer} + E_{fryer} + E_{freezer} \tag{I.7}$$

$$E_{totalsaved} = 4.561 \cdot 10^4 \text{ W}$$

The above calculations were done for 1 000 kg/h at inspection line 1. Currently, production is run at approximately 7 500 kg/hr in. This gives a total of 5482.5 kg/h. Thus, the total energy that is saved is:

$$E_{currentsaved} := \left(\frac{5482.5}{1000}\right) \cdot E_{totalsaved} \tag{I.8}$$

$$E_{currentsaved} = 2.5 \cdot 10^5 \text{ W}$$

Costing in terms of coal:

$$E_{totslcoalsaved} = \left(\frac{5482.5}{1000}\right) \cdot \left(E_{blancher} + E_{dryer} + E_{fryer}\right)$$

$$E_{totslcoalsaved} = 240.188 kW \tag{I.9}$$

Costing in terms of electricity with coefficient of performance, COP = 2.5:

$$E_{totalelec} = \left(\frac{5482.5}{1000}\right) \cdot \frac{E_{freezer}}{2.5}$$

$$E_{totalelec} = 3931.9 W \tag{I.10}$$

Currently, the cost for coal is R0.0157/MJ and for electricity R0.0446/MJ. Assuming the plant is operating 24 hours per day and 340 days per year, the savings per year would be:

$Coal_{cost} := \frac{0.0157}{10^6 \text{ J}}$	$Elec_{cost} := \frac{0.0446}{10^6 \text{ J}}$

$Savings_{coal} = E_{totalcoalsaved} \cdot 340days \cdot 24h \cdot 3600s \cdot Coal_{\cos t}$	$Savings_{coal} = R110801.30$
$Savings_{elec} = E_{totalelec} \cdot 340days \cdot 24h \cdot 3600s \cdot Elec_{\cos t}$	$Savings_{elec} = R5151.46$

Table I. 5: Energy savings calculation table

The salary savings due to nine workers replaced by the optical sorter and assuming an hourly rate of R8 and two 12-hour shifts for 340 days, are as follows:

$$\text{Salary}_{\text{savings}} := 9 \cdot 8 \cdot 24 \cdot 340 \qquad \text{(I.11)}$$

$$\text{Salary}_{\text{savings}} = 5.875 \cdot 10^{5}$$

Salary savings of R587 500 per annum is possible. Thus, the total savings from the optical sorter are:

$$Savings_{total} = Salary_{savings} + Savings_{coal} + Savings_{elec}$$
$$Savings_{total} = R703452.76 \qquad \text{(I.12)}$$

For the optical sorter, one needs to budget 10% of the gross value for maintenance for the first two years and thereafter 20%.

Cost of Optical Sorter

$$Cost_{total} = R1300000.00 \qquad \text{(I.13)}$$

Maintenance for the first two years at 10% of the gross value:

$$\text{Cost}_{\text{maintenance}} := 0.1 \cdot \text{Cost}_{\text{opticalsorter}}$$

Payback period:

$$n = \frac{\ln\left(\frac{1}{1-\frac{1.3\cdot 10^{6}}{573452.76}\cdot 0.17}\right)}{\ln(1+0.17)} \qquad \text{(I.14)}$$

$$n = 3.1003$$

Thus, in 2.7 years (35 months) the optical sorter will be paid in full.

Savings Realised Through Isolated (Shortened) Sorting Line

By eliminating the sorting line, it is possible to decrease the temperature increase of the final product by isolating the whole area and ensuring a shorter exposure time of fries to the environment. Let us say that temperature increase and exposure time are directly proportional and that by isolating the area around the fries, it is possible to get the environment temperature down to 0 ^{O}C – 5 ^{O}C. Therefore, the final mass flow of 3 425 kg/h frozen fries will experience a decrease in temperature increase from -6 ^{O}C to -14 ^{O}C. This will result in savings of:

$m_{finished} := 3425 \frac{kg}{hr}$	$\Delta T_{decrease} := (-6 + 14)\ K$
$E_{finishedproduct} := c_{ppotato} \cdot m_{finished} \cdot \Delta T_{decrease}$	$E_{finishedproduct} = 2.877 \cdot 10^{4}\ W$
$Savings_{finishedyear} := 340\ day \cdot E_{finishedproduct} \cdot Elec_{cost}$	$Savings_{finishedyear} = 3.769 \cdot 10^{4}$

Table I. 6: Sorting line savings

From the above calculation, it was estimated that R37 690 can be saved annually on reduced electricity cost due to reduced heat gain of the frozen product.

J Appendix: Dryer Test Measurements and Calculations

J.1 Dryer Test Measurements

Section	Dry-bulb	Wet-bulb	T_{Fries}
Outside plant	15.8	13.3	
Inside plant	20.7	17.0	
	21.4	17.0	
6 bo	82.5	35.1	21.6
	74.6	32.1	
6 onder	25.6	23.8	
	25.2	23.6	
5 onder	37.1	28.8	32.5
	36.5	28.0	
5 bo	39.7	33.8	
	39.1	33.5	
4 bo	62.6	37.7	37.2
	63.9	37.9	
4 onder	45.0	37.8	
	42.3	38.4	
3 onder	78.6	47.2	44.3
	79.2	47.4	
3 bo	52.5	43.1	
	52.7	43.1	
2 bo	74.4	43.8	44.0
	77.6	44.8	
2 onder	53.6	44.0	
	53.6	44.9	
1 onder	71.7	44.9	33.7
	75.5	45.6	
1 bo	39.5	34.7	
	36.4	31.4	

Table J. 1: Dryer test measurements

J.2 Dryer Test Calculations

Outside air			Evaporation enthalpy of water
Tdb	Tnb	Patm	ifgwo
21.100	15.100	101325	2501600

Section	Tdb	[K]	Tnb	[K]	Pz1 [Pa]	Pz2 [Pa]
Outside	21.1	294.25	15.1	288.25	2501.057	1715.143
1.000	35.0	308.15	28.4	301.55	5623.136	3868.164
1.500	62.7	335.85	39.2	312.35	22551.243	7068.242
2.000	67.7	340.85	38.4	311.55	28195.073	6770.582
2.500	46.8	319.95	37.4	310.55	10508.034	6413.792
3.000	48.9	322.05	36.4	309.55	11680.814	6073.383
3.500	62.0	335.15	35.1	308.25	21843.660	5654.308
4.000	45.3	318.45	30.2	303.35	9733.632	4291.544
4.500	36.3	309.45	30.6	303.75	6040.219	4390.917
5.000	34.5	307.65	28.9	302.05	5469.497	3981.955
5.500	32.6	305.75	24.2	297.35	4918.357	3018.806
6.000	86.6	359.75	34.9	308.05	61531.893	5592.112
6.500	31.4	304.55	25.0	298.15	4595.677	3166.717

ωz2	cpa1	cpa2	cpv1	cpv2	ha1	ha2
0.011	1006.762	1006.597	1880.646	1875.421	21242.68	15199.61
0.025	1007.275	1007.009	1893.962	1887.368	35254.64	28599.07
0.047	1008.822	1007.465	1931.543	1898.498	63253.11	39492.64
0.045	1009.173	1007.428	1940.923	1897.611	68321.01	38685.23
0.042	1007.850	1007.382	1907.573	1896.517	47167.38	37676.09
0.040	1007.966	1007.337	1910.313	1895.441	49289.51	36667.06
0.037	1008.774	1007.280	1930.310	1894.066	62544.00	35355.52
0.028	1007.770	1007.078	1905.682	1889.110	45651.98	30413.75
0.028	1007.332	1007.094	1895.334	1889.502	36566.17	30817.06
0.025	1007.254	1007.028	1893.441	1887.848	34750.26	29103.11
0.019	1007.174	1006.861	1891.495	1883.441	32833.87	24366.04
0.036	1010.694	1007.271	1987.537	1893.857	87526.14	35153.76
0.020	1007.125	1006.888	1890.293	1884.176	31623.73	25172.20

hv1	hv2	cpw	νw	hf	ω	PH	hav	Mass of fries	T_{fries}
2541281.63	2529918.85	4189.022	0.001001	63355.65	0.008222	0.528545	42137.35	0.150	24.0
2567888.65	2555201.25	4178.007	0.001004	118757.11	0.021842	0.611307	91343.54	0.154	
2622707.75	2576021.13	4175.412	0.001007	163778.24	0.036095	0.246435	157919.16		
2633000.47	2574468.25	4175.432	0.001007	160438.64	0.031498	0.173215	151255.76	0.151	37.4
2590874.42	2572529.74	4175.493	0.001007	156265.46	0.037818	0.552668	145147.89		
2595014.30	2570594.05	4175.596	0.001006	152093.68	0.034096	0.450798	137769.73	0.152	35.8
2621279.22	2568081.72	4175.793	0.001006	146672.28	0.024984	0.179127	128034.16		
2587927.39	2558651.12	4177.208	0.001004	126253.46	0.020992	0.339854	99978.09		30.4
2570400.63	2559418.77	4177.052	0.001004	127919.57	0.025695	0.665487	102612.24		
2566923.70	2556158.81	4177.770	0.001004	120839.27	0.023023	0.661240	93848.98		24.9
2563262.75	2547179.28	4180.486	0.001003	101269.36	0.015536	0.502041	72657.48		
2673720.68	2567695.61	4175.830	0.001006	145838.40	0.014092	0.036480	125202.96		
2560955.21	2548704.39	4179.945	0.001003	104600.26	0.017340	0.597977	76030.71		

Table J. 2: Spreadsheet test calculations for dryer

K Appendix: Visual Basic Program for Heat Exchanger Analysis

K.1 Heat Exchanger Code

```
Option Explicit

Dim i As Integer
Dim j As Integer
Dim m As Integer
Dim pi
Dim Nn
'Segment flow conditions:
Dim dio As Double
Dim mw As Double, kss
Dim txtri As Double
Dim txtAe As Double
Dim txtAp As Double
Dim txtmse As Double
Dim txtmwe As Double
Dim tw As Double
Dim dx As Double
Dim eqBend As Double
'Subprocedure for calculations
Dim Tsin(0 To 100, 0 To 100), Twin(0 To 100, 0 To 100), Hsin(0 To 100, 0 To 100), Hwin(0
To 100, 0 To 100)
Dim rhos(0 To 100, 0 To 100), kviscs(0 To 100, 0 To 100), ks(0 To 100, 0 To 100), Prs(0 To
100, 0 To 100)
Dim Res(0 To 100, 0 To 100), Nuds(0 To 100, 0 To 100), hcs(0 To 100, 0 To 100), rhow(0
To 100, 0 To 100)
Dim kviscw(0 To 100, 0 To 100), dviscs(0 To 100, 0 To 100), kw(0 To 100, 0 To 100), Prw(0
To 100, 0 To 100), Rew(0 To 100, 0 To 100)
Dim f(0 To 100, 0 To 100), Nudw(0 To 100, 0 To 100), hcw(0 To 100, 0 To 100), term1(0 To
100, 0 To 100)
Dim term3(0 To 100, 0 To 100), UP(0 To 100, 0 To 100), deltaQ(0 To 100, 0 To 100),
Hsout(0 To 100, 0 To 100)
Dim Hwout(0 To 100, 0 To 100), Tsout(0 To 100, 0 To 100), Twout(0 To 100, 0 To 100),
deltaPs(0 To 100, 0 To 100)
Dim Twt(0 To 100, 0 To 100), fdrag(0 To 100, 0 To 100), deltaPw(0 To 100, 0 To 100),
Rys(0 To 100, 0 To 100), Kloss(0 To 100, 0 To 100)
Dim term2, msCond(0 To 100, 0 To 100), msCondTotal(0 To 100, 0 To 100), mse(0 To 100,
0 To 100), mseOut(0 To 100, 0 To 100)
Dim bFin(0 To 100, 0 To 100), zFin(0 To 100, 0 To 100), min2epowerzFin(0 To 100, 0 To
100), tanhzFin(0 To 100, 0 To 100)
Dim velFree(0 To 100, 0 To 100), velMax(0 To 100, 0 To 100), deltaPwBend(0 To 100, 0 To
100), deltaPwPrevious
Dim Atotal
Dim Nrow As Integer, numRow, numCol, numPipe, numFin

Dim ibegin As Integer
Dim iend As Integer
```

```
Dim iStart As Integer
Dim iFinish As Integer
Dim deltaPsTotal(0 To 100, 0 To 100)
Dim deltaPwTotal(0 To 100, 0 To 100)
Dim deltaQTotal(0 To 100, 0 To 100)
Dim deltaAtotal, deltaAtotalPrevious
Dim mseTotalExit
Dim betaFin(0 To 100, 0 To 100)
Dim thetaFin(0 To 100, 0 To 100)
Dim term3Fin(0 To 100, 0 To 100)
Dim thetaFintotal(0 To 100, 0 To 100)
Dim UFins(0 To 100, 0 To 100)



Private Sub cmdcalculate_Click()
'Segment flow conditions:
pi = 3.141592654
dio = Val(txtdo.Text)
mw = Val(txtmw.Text)
numCol = Val(txtNrc.Text)
numRow = Val(txtNrows.Text)
numPipe = Val(txtnumPipes.Text)
widthHE = Val(txtWidth.Text)
kss = Val(txtkss.Text)
RelRough = Val(txtRelRough.Text)
dx = Val(txtls.Text)
numFin = 0
tw = Val(txttw.Text)
At = widthHE * widthHE

eqBend = 1.2
txtri = dio / 2 - tw
numElem = numCol * numPipe

txtAe = At / numElem
txtAp = pi * txtri ^ 2
txtmse = txtms / numElem
txtmwe = mw / numPipe


ibegin = 0
iend = widthHE / dx - 1

Twin(0, 0) = Val(txtTwin.Text)

For i = ibegin To iend
   Tsin(i, 0) = Val(txtTsin.Text)
   mse(i, 0) = txtmse
Next i

m = numRow
```

```
iStart = ibegin
iFinish = iend
parameterj = -1
deltaPsPrevious = 0
deltaPwPrevious = 0
deltaQPrevious = 0
deltaAtotalPrevious = 0
msCondPrevious = 0

For j = 0 To m

   parameterj = -parameterj
   For i = ibegin To iend Step parameterj
      Nrow = 0
      Call CalculationFins(i, j, Twin(), Tsin(), Twout(), Tsout(), deltaQ(), deltaPs(), deltaPw(),
deltaPwBend(), Atotal)

      If parameterj = 1 Then
         Twin(i + 1, j) = Twout(i, j)
      Else
          If i - 1 >= iend Then Twin(i - 1, j) = Twout(i, j)
         End If
      Tsin(i, j + 1) = Tsout(i, j)
      mse(i, j + 1) = mseOut(i, j)

      'Pressurefall limit for water in heat exchanger
      deltaPwTotal(i, j) = deltaPw(i, j) + deltaPwPrevious
      deltaPwPrevious = deltaPwTotal(i, j)
      msCondTotal(i, j) = msCond(i, j) + msCondPrevious
      msCondPrevious = msCondTotal(i, j)

      'Heat transfer summation
      deltaQTotal(i, j) = deltaQ(i, j) + deltaQPrevious
      deltaQPrevious = deltaQTotal(i, j)
      deltaAtotal = deltaAtotalPrevious + Atotal
      deltaAtotalPrevious = deltaAtotal

      Next i
      If parameterj = -1 Then
         deltaPwPrevious = deltaPwTotal(iStart, j) + deltaPwBend(iStart, j)
         Else
         deltaPwPrevious = deltaPwTotal(iFinish, j) + deltaPwBend(iFinish, j)
         End If
   Twin(iend, j + 1) = Twout(iend, j)

   ibeginnew = ibegin
   ibegin = iend
   iend = ibeginnew

   'Pressurefall limit for steam in heat exchanger
   deltaPsTotal(iStart, j) = deltaPs(iStart, j) + deltaPsPrevious
   deltaPsPrevious = deltaPsTotal(iStart, j)
```

```
Next j

For j = 0 To m
For i = 0 To iFinish
   ListTwin.AddItem Format(Twin(i, j), "0.000000")
   ListTsout.AddItem Format(Tsin(i, j), "0.000000")
   ListdeltaQ.AddItem Format(deltaQ(i, j), "0.000000")
   ListdeltaPs.AddItem Format(deltaPs(i, j), "0.000000")
   ListdeltaPw.AddItem Format(deltaPw(i, j), "0.000000")

Next i
Next j

msePrevious = 0
For i = 0 To iFinish
mseTotalExit = mse(i, m) + msePrevious
msePrevious = mseTotalExit
Next i

End Sub



Private Sub CmdCalculationFins_Click()

'Segment flow conditions:
pi = 3.141592654
dio = Val(txtdo.Text)
mw = Val(txtmw.Text)
numCol = Val(txtNrc.Text)
numRow = Val(txtNrows.Text)
numPipe = Val(txtnumPipes.Text)
widthHE = Val(txtWidth.Text)
kss = Val(txtkss.Text)
RelRough = Val(txtRelRough.Text)
dx = Val(txtls.Text)
numFin = Val(txtnumFins.Text)
tw = Val(txttw.Text)
At = widthHE * widthHE

eqBend = 1.2
txtri = dio / 2 - tw
numElem = numCol * numPipe

txtAe = At / numElem
txtAp = pi * txtri ^ 2
txtmse = txtms / numElem
txtmwe = mw / numPipe


ibegin = 0
```

```
iend = widthHE / dx - 1

Twin(0, 0) = Val(txtTwin.Text)

For i = ibegin To iend
   Tsin(i, 0) = Val(txtTsin.Text)
   mse(i, 0) = txtmse
Next i

m = numRow
iStart = ibegin
iFinish = iend
parameterj = -1
deltaPsPrevious = 0
deltaPwPrevious = 0
deltaQPrevious = 0
deltaAtotalPrevious = 0
msCondPrevious = 0

For j = 0 To m

   parameterj = -parameterj
   For i = ibegin To iend Step parameterj
      Nrow = 0
      Call CalculationFins(i, j, Twin(), Tsin(), Twout(), Tsout(), deltaQ(), deltaPs(), deltaPw(),
deltaPwBend(), Atotal)

      If parameterj = 1 Then
         Twin(i + 1, j) = Twout(i, j)
      Else
          If i - 1 >= iend Then Twin(i - 1, j) = Twout(i, j)
         End If
      Tsin(i, j + 1) = Tsout(i, j)
      mse(i, j + 1) = mseOut(i, j)

      'Pressurefall limit for water in heat exchanger
      deltaPwTotal(i, j) = deltaPw(i, j) + deltaPwPrevious
      deltaPwPrevious = deltaPwTotal(i, j)
      msCondTotal(i, j) = msCond(i, j) + msCondPrevious
      msCondPrevious = msCondTotal(i, j)

      'Heat transfer summation
      deltaQTotal(i, j) = deltaQ(i, j) + deltaQPrevious
      deltaQPrevious = deltaQTotal(i, j)
      deltaAtotal = deltaAtotalPrevious + Atotal
      deltaAtotalPrevious = deltaAtotal

   Next i
      If parameterj = -1 Then
         deltaPwPrevious = deltaPwTotal(iStart, j) + deltaPwBend(iStart, j)
         Else
         deltaPwPrevious = deltaPwTotal(iFinish, j) + deltaPwBend(iFinish, j)
```

```
        End If
    Twin(iend, j + 1) = Twout(iend, j)

    ibeginnew = ibegin
    ibegin = iend
    iend = ibeginnew

    'Pressurefall limit for steam in heat exchanger
    deltaPsTotal(iStart, j) = deltaPs(iStart, j) + deltaPsPrevious
    deltaPsPrevious = deltaPsTotal(iStart, j)

Next j

For j = 0 To m
For i = 0 To iFinish
    ListTwin.AddItem Format(Twin(i, j), "0.000000")
    ListTsout.AddItem Format(Tsin(i, j), "0.000000")
    ListdeltaQ.AddItem Format(deltaQ(i, j), "0.000000")
    ListdeltaPs.AddItem Format(deltaPs(i, j), "0.000000")
    ListdeltaPw.AddItem Format(deltaPw(i, j), "0.000000")

Next i
Next j

msePrevious = 0
For i = 0 To iFinish
mseTotalExit = mse(i, m) + msePrevious
msePrevious = mseTotalExit
Next i

End Sub




Private Sub cmdCombi_Click()
'Segment flow conditions:
pi = 3.141592654
dio = Val(txtdo.Text)
mw = Val(txtmw.Text)
numCol = Val(txtNrc.Text)
numRow = Val(txtNrows.Text)
numPipe = Val(txtnumPipes.Text)
widthHE = Val(txtWidth.Text)
kss = Val(txtkss.Text)
RelRough = Val(txtRelRough.Text)
dx = Val(txtls.Text)
numFin = Val(txtnumFins.Text)
tw = Val(txttw.Text)
At = widthHE * widthHE
```

```
eqBend = 1.2
txtri = dio / 2 - tw
numElem = numCol * numPipe

txtAe = At / numElem
txtAp = pi * txtri ^ 2
txtmse = txtms / numElem
txtmwe = mw / numPipe


ibegin = 0
iend = widthHE / dx - 1

Twin(0, 0) = Val(txtTwin.Text)

For i = ibegin To iend
   Tsin(i, 0) = Val(txtTsin.Text)
   mse(i, 0) = txtmse
Next i

m = numRow
iStart = ibegin
iFinish = iend
parameterj = -1
deltaPsPrevious = 0
deltaPwPrevious = 0
deltaQPrevious = 0
deltaAtotalPrevious = 0
msCondPrevious = 0

For j = 0 To 9

   parameterj = -parameterj
   For i = ibegin To iend Step parameterj
      Nrow = 0
      Call CalculationFins(i, j, Twin(), Tsin(), Twout(), Tsout(), deltaQ(), deltaPs(), deltaPw(),
deltaPwBend(), Atotal)

      If parameterj = 1 Then
         Twin(i + 1, j) = Twout(i, j)
      Else
          If i - 1 >= iend Then Twin(i - 1, j) = Twout(i, j)
         End If
      Tsin(i, j + 1) = Tsout(i, j)
      mse(i, j + 1) = mseOut(i, j)

      'Pressurefall limit for water in heat exchanger
      deltaPwTotal(i, j) = deltaPw(i, j) + deltaPwPrevious
      deltaPwPrevious = deltaPwTotal(i, j)
      msCondTotal(i, j) = msCond(i, j) + msCondPrevious
      msCondPrevious = msCondTotal(i, j)
```

```
        'Heat transfer summation and heat transfer area summation
        deltaQTotal(i, j) = deltaQ(i, j) + deltaQPrevious
        deltaQPrevious = deltaQTotal(i, j)
        deltaAtotal = deltaAtotalPrevious + Atotal
        deltaAtotalPrevious = deltaAtotal

        Next i
        If parameterj = -1 Then
           deltaPwPrevious = deltaPwTotal(iStart, j) + deltaPwBend(iStart, j)
           Else
           deltaPwPrevious = deltaPwTotal(iFinish, j) + deltaPwBend(iFinish, j)
           End If
    Twin(iend, j + 1) = Twout(iend, j)

    ibeginnew = ibegin
    ibegin = iend
    iend = ibeginnew

    'Pressurefall limit for steam in heat exchanger
    deltaPsTotal(iStart, j) = deltaPs(iStart, j) + deltaPsPrevious
    deltaPsPrevious = deltaPsTotal(iStart, j)

Next j
ibegin = iStart
iend = iFinish
parameterj = -1
For j = 10 To m
numFin = 0
    parameterj = -parameterj
    For i = ibegin To iend Step parameterj

        Call CalculationFins(i, j, Twin(), Tsin(), Twout(), Tsout(), deltaQ(), deltaPs(), deltaPw(),
deltaPwBend(), Atotal)

        If parameterj = 1 Then
           Twin(i + 1, j) = Twout(i, j)
        Else
            If i - 1 >= iend Then Twin(i - 1, j) = Twout(i, j)
           End If
        Tsin(i, j + 1) = Tsout(i, j)
        mse(i, j + 1) = mseOut(i, j)

        'Pressurefall limit for water in heat exchanger
        deltaPwTotal(i, j) = deltaPw(i, j) + deltaPwPrevious
        deltaPwPrevious = deltaPwTotal(i, j)
        msCondTotal(i, j) = msCond(i, j) + msCondPrevious
        msCondPrevious = msCondTotal(i, j)

        'Heat transfer summation
        deltaQTotal(i, j) = deltaQ(i, j) + deltaQPrevious
        deltaQPrevious = deltaQTotal(i, j)
        deltaAtotal = deltaAtotalPrevious + Atotal
```

```
        deltaAtotalPrevious = deltaAtotal

        Next i
        If parameterj = -1 Then
            deltaPwPrevious = deltaPwTotal(iStart, j) + deltaPwBend(iStart, j)
            Else
            deltaPwPrevious = deltaPwTotal(iFinish, j) + deltaPwBend(iFinish, j)
            End If
    Twin(iend, j + 1) = Twout(iend, j)

    ibeginnew = ibegin
    ibegin = iend
    iend = ibeginnew

    'Pressurefall limit for steam in heat exchanger
    deltaPsTotal(iStart, j) = deltaPs(iStart, j) + deltaPsPrevious
    deltaPsPrevious = deltaPsTotal(iStart, j)

Next j
For j = 0 To m
For i = 0 To iFinish
    ListTwin.AddItem Format(Twin(i, j), "0.000000")
    ListTsout.AddItem Format(Tsin(i, j), "0.000000")
    ListdeltaQ.AddItem Format(deltaQ(i, j), "0.000000")
    ListdeltaPs.AddItem Format(deltaPs(i, j), "0.000000")
    ListdeltaPw.AddItem Format(deltaPw(i, j), "0.000000")

Next i
Next j

msePrevious = 0
For i = 0 To iFinish
mseTotalExit = mse(i, m) + msePrevious
msePrevious = mseTotalExit
Next i
End Sub

Private Sub cmddisplay_Click()
picPlot.Print deltaPsTotal(iStart, m), "Total steam pressure fall"
picPlot.Print deltaPwTotal(iend, m), "Total water pressure fall"
picPlot.Print deltaQTotal(iFinish, m), "Total heat transfer"
picPlot.Print deltaAtotal, "Total heat transfer surface (m2)"

picPlot.Print msCondTotal(iFinish, m), "Total condensate per column"
picPlot.Print mseTotalExit, "Total steam flow at exit"
picPlot.Print (mseTotalExit + msCondTotal(iFinish, m)) * 14, "total water out"


End Sub
```

```
Private Sub cmdok_Click()
End
End Sub



Private Sub CalculationFins(i, j, Twin(), Tsin(), Twout(), Tsout(), deltaQ(), deltaPs(),
deltaPw(), deltaPwBend(), Atotal)

'Calculation of Overall heat transfer coefficient, UA
'Calculation of convective heat transfer coefficient for the steam side:

   Hsin(i, j) = mse(i, j) * (1986.3 * Tsin(i, j) + 1935600) + msCond(i, j) * hfs   '(Correlation
obtain from Schmidt,  1969)
   Hwin(i, j) = txtmwe * (4185.1 * Twin(i, j) - 1143000)        '(Correlation obtain from
Schmidt,  1969)
'______________________________________________________________________________________
__________________________________
'Steam properties at txtTsin:
rhos(i, j) = 3.4503922053E-06 * Tsin(i, j) ^ 2 - 0.00420531051 * Tsin(i, j) + 1.6780736016
kviscs(i, j) = 0.000000146042 * Tsin(i, j) - 0.00003422741
ks(i, j) = 0.0000837273 * Tsin(i, j) - 0.0066655591
Prs(i, j) = -0.0002418182 * Tsin(i, j) + 1.073053
Res(i, j) = (mse(i, j) * dio) / (rhos(i, j) * txtAe * kviscs(i, j))
If Res(i, j) < 10000 Then
   Nuds(i, j) = 0.3 + (0.62 * (Res(i, j) ^ 0.5) * Prs(i, j) ^ (1 / 3)) / ((1 + (0.4 / Prs(i, j)) ^ (2 / 3))
^ (1 / 4))
   ElseIf 10000 <= Res(i, j) < 400000 Then
   Nuds(i, j) = 0.3 + (((0.62 * (Res(i, j) ^ 0.5) * Prs(i, j) ^ (1 / 3)) / (1 + (0.4 / Prs(i, j)) ^ (2 /
3)) ^ (1 / 4))) * (1 + (Res(i, j) / 282000) ^ 0.5)
   ElseIf Res(i, j) > 400000 Then
   Nuds(i, j) = 0.3 + (((0.62 * (Res(i, j) ^ 0.5) * Prs(i, j) ^ (1 / 3)) / (1 + (0.4 / Prs(i, j)) ^ (2 /
3)) ^ (1 / 4))) * ((1 + (Res(i, j) / 282000) ^ (5 / 8)) ^ 0.8)
End If
'Convection heat transfer coefficient for steam
   rhol = 957.8544
   rhov = 0.59776
   g = 9.81
   ksl = 0.681
   mul = 279 * 10 ^ (-6)
   hfs = 419040
   hgs = 2676100
   hfgs = 2257000

If Tsin(i, j) > 373.15 Then
   hcs(i, j) = ks(i, j) * Nuds(i, j) / dio
   Else
   Nrow = Nrow + 1
   hcs(i, j) = 0.728 * (((rhol - rhov) * g * hfgs * (ksl) ^ 3) / (Nrow * (mul / rhol) * dio *
(Tsin(i, j) - Twin(i, j)))) ^ 0.25
   End If
```

```
'________________________________________________________________________________
________________________________
'Water properties at txtTwin:
rhow(i, j) = -0.0038388230179 * Twin(i, j) ^ 2 + 2.0464231519 * Twin(i, j) + 727.82179247
kviscw(i, j) = (-2.83876495726479E-12 * Twin(i, j) ^ 3 + 2.92563296346138E-09 * Twin(i, j) ^ 2 - 1.00963920098292E-06 * Twin(i, j) + 1.1708641282173E-04)
kw(i, j) = 0.3765493686706 * Log(Twin(i, j)) - 1.5368285999447
Prw(i, j) = 0.0017696969697 * Twin(i, j) ^ 2 - 1.2346642424 * Twin(i, j) + 217.42872345
Rew(i, j) = (txtmwe * txtdo) / (rhow(i, j) * txtAp * kviscw(i, j))
f(i, j) = (0.79 * Log(Rew(i, j)) - 1.64) ^ -2
If Rew(i, j) < 3000 Then
   Nudw(i, j) = 4.364
   ElseIf 3000 <= Rew(i, j) < 10000 Then
   Nudw(i, j) = ((f(i, j) / 8) * (Rew(i, j) - 1000) * (Prw(i, j))) / (1 + 12.7 * ((f(i, j) / 8) ^ 0.5) * (Prw(i, j) ^ (2 / 3) - 1))
   ElseIf Rew(i, j) > 10000 Then
   Nudw(i, j) = 0.023 * (Rew(i, j) ^ 0.8) * (Prw(i, j) ^ 0.4)
End If
'Convection heat transfer coefficient for water
hcw(i, j) = kw(i, j) * Nudw(i, j) / (2 * txtri)
'________________________________________________________________________________
________________________________
'Overall heat transfer coefficients:
'Fin efficiency
Call FinEfficiency(hcs(), thetaFin(), thetaFintotal(), Atotal)

term1(i, j) = Atotal / (hcw(i, j) * 2 * pi * txtri)
term2 = Atotal * (Log(dio / (2 * txtri))) / (2 * pi * kss * tw)
term3Fin(i, j) = 1 / (hcs(i, j) * thetaFintotal(i, j))
UFins(i, j) = (term1(i, j) + term2 + term3Fin(i, j)) ^ -1

deltaQ(i, j) = UFins(i, j) * Atotal * (Tsin(i, j) - Twin(i, j))
'________________________________________________________________________________
________________________________
Hsout(i, j) = Hsin(i, j) - deltaQ(i, j)
Hwout(i, j) = Hwin(i, j) + deltaQ(i, j)
If Tsin(i, j) > 373.15 Then
   Tsout(i, j) = (Hsout(i, j) / mse(i, j) - 1935600) / 1986.3
   msCond(i, j) = 0
   Else
   Tsout(i, j) = 373.15
   msCond(i, j) = deltaQ(i, j) / hfgs
   End If

mseOut(i, j) = mse(i, j) - msCond(i, j)
Twout(i, j) = (Hwout(i, j) / txtmwe + 1143000) / 4185.1
'________________________________________________________________________________
________________________________
'Steam pressure fall calculations fins included
'freestream loss coefficient
Afrontal = pi * dx * dio
dviscs(i, j) = kviscs(i, j) * rhos(i, j)
```

```
velFree(i, j) = mse(i, j) / (rhos(i, j) * Afrontal)
st = 0.05
velMax(i, j) = velFree(i, j) * (st / (st - dio))
deltaPs(i, j) = (j + 1) * rhos(i, j) * velMax(i, j) ^ 2 * 0.5

'Water pressure drop calculations
Twt(i, j) = deltaQ(i, j) / hcw(i, j) * txtAp ^ 2 + Twin(i, j)
fdrag(i, j) = 0.3086 * ((Log(6.9 / Rew(i, j))) / Log(10) + (RelRough / 3.75) ^ 1.11) ^ (-2)
deltaPw(i, j) = fdrag(i, j) * dx ^ 2 * mw ^ 2 / (2 * txtri * rhow(i, j) * txtAp ^ 2)
deltaPwBend(i, j) = fdrag(i, j) * eqBend ^ 2 * mw ^ 2 / (2 * txtri * rhow(i, j) * txtAp ^ 2)
End Sub


Private Function FinEfficiency(hcs(), thetaFin(), thetaFintotal(), Atotal)

'Fin properties
tFin = Val(txtfinThick.Text)
kFin = kss
bFin(i, j) = (hcs(i, j) / (0.5 * tFin * kFin)) ^ 0.5
r2Fin = Val(txtfindia.Text) / 2
r1Fin = dio / 2
phiFin = (r2Fin / r1Fin - 1) * (1 + 0.35 * Log(r2Fin / r1Fin))
zFin(i, j) = bFin(i, j) * dio * phiFin / 2

SFin = 2 * pi * (r2Fin ^ 2 - r1Fin ^ 2)
AFin = 2 * SFin
Atotal = dx * pi * dio + AFin * numFin
VFin = 0.5 * SFin * tFin

'Calculation of hyperbolic Tan:
min2epowerzFin(i, j) = 1 + (-2 * zFin(i, j)) + (-2 * zFin(i, j)) ^ 2 / 2 + (-2 * zFin(i, j)) ^ 3 / 6 +
(-2 * zFin(i, j)) ^ 4 / 24 + (-2 * zFin(i, j)) ^ 5 / 120
tanhzFin(i, j) = (1 - min2epowerzFin(i, j)) / (1 + min2epowerzFin(i, j))
thetaFin(i, j) = tanhzFin(i, j) / zFin(i, j)
If numFin >= 1 Then
thetaFintotal(i, j) = 1 - AFin * (1 - thetaFin(i, j)) / Atotal
Else
thetaFintotal(i, j) = 1
End If

End Function


Private Sub cmdPlot_Click()
Dim bo As Double
Dim regs As Double
Dim onder As Double
Dim links As Double
Dim yMaks As Double
Dim xMaks As Double
Dim yDelta As Double
Dim xDelta As Double
```

```
Dim yMin As Double
Dim xMin As Double

Dim X As Double
Dim Y As Double

    yMaks = -273
    yMin = 99999
    For X = 1 To ListTwin.ListCount - 1
        If yMaks < ListTwin.List(X) Then yMaks = ListTwin.List(X)
        If yMin > ListTwin.List(X) Then yMin = ListTwin.List(X)
        Next X

    xMaks = ListTwin.ListCount
    xMin = 0

    yDelta = yMaks - yMin
    xDelta = xMaks - xMin
    '   Kry maksimum y waarde
    bo = yMaks + 0.05 * yDelta
    '   Kry minimum y waarde
    onder = yMin - 0.05 * yDelta
    '   Kry maksimum x waarde
    regs = xMaks + 0.05 * xDelta
    '   Kry minimum x waarde
    links = xMin - 0.05 * xDelta

    ' Create plotting window
    picPlot.Scale (links, bo)-(regs, onder)
    picPlot.PSet (xMin, yMin)
    picPlot.Print "Plot of water temperature"

    picPlot.Line (xMin, yMin)-(xMin, yMaks)
    picPlot.Line (xMin, yMin)-(xMaks, yMin)
    picPlot.FillColor = vbBlue
    picPlot.FillStyle = vbFSSolid

    For X = 1 To ListTwin.ListCount - 1
        ' picPlot.Line (x, ListTwin.List(x))-(x + 1, ListTwin.List(x + 1))
       'picPlot.PSet (x, ListTwin.List(x)), vbRed
        picPlot.Circle (X, ListTwin.List(X)), 0.08, vbBlue
        'picPlot.Circle (X, ListTsout.List(X)), 0.08, vbBlue

        Next X

    End Sub


Private Sub cmdSave_Click()

    Open App.Path & "\pressurefallwater.txt" For Output As #1
```

```
    parameterj = -1
    ibegin = 0
    iend = iFinish
    For j = 0 To m
        parameterj = -parameterj
        For i = ibegin To iend Step parameterj
        Write #1, i, j, deltaPwTotal(i, j), deltaPsTotal(i, j), deltaQTotal(i, j), hcs(i, j), hcw(i, j),
Tsin(i, j), Twin(i, j)
        Next i
        ibeginnew = ibegin
        ibegin = iend
        iend = ibeginnew
        Next j
    Close #1

    Open App.Path & "\temperaturewater.txt" For Output As #2
    Dim k As Integer
    For k = 0 To ListTwin.ListCount - 1
        Write #2, k, ListTwin.List(k), ListTsout.List(k)
        Next k
    Close #2

    Open App.Path & "\temperaturesteam.txt" For Output As #3
    For j = 0 To m - 1
        Write #3, j, Tsin(iStart, j)
        Next j
    Close #3

    Open App.Path & "\pressurefallsteam.txt" For Output As #4
    For j = 0 To m - 1
        Write #4, j, deltaPsTotal(iStart, j)
        Next j
    Close #4
    End Sub




Private Sub picPlot_Click()
    picPlot.Cls
    End Sub
```

K.2 Heat Exchanger Program Form

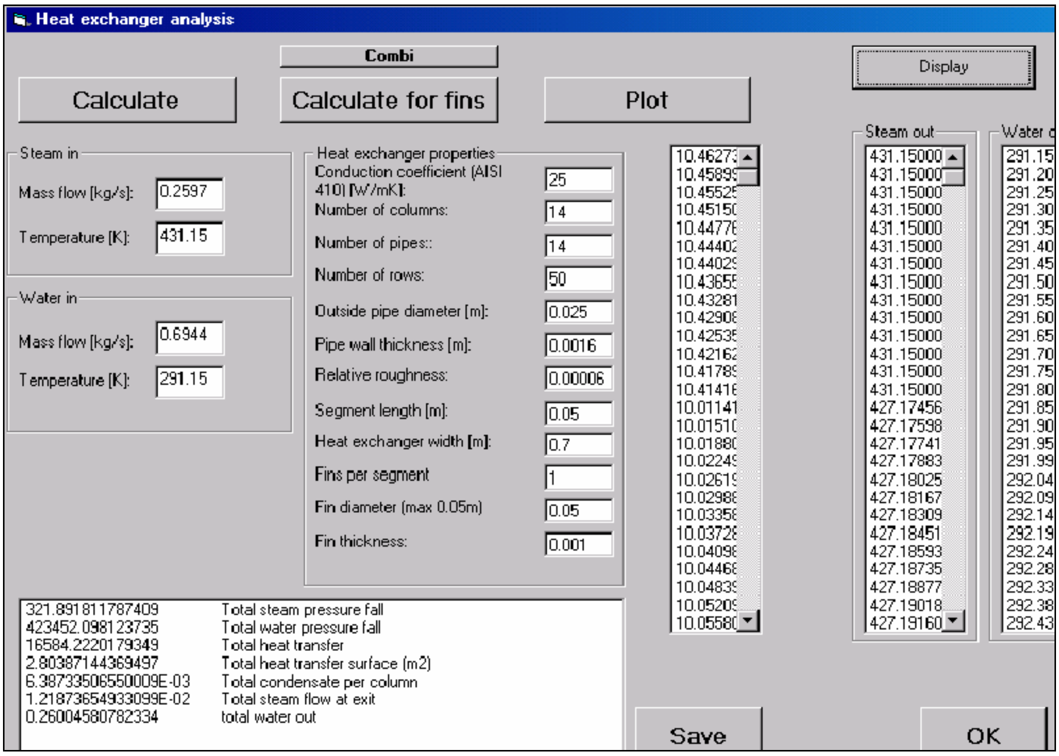


Figure K. 1: Heat exchanger program form

K.3 Correlations for Steam and Water Properties

Density (kg/m^3), specific heat (kJ/kgK), dynamic viscosity (m^2/s), thermal conductivity coefficient (W/mK) and the Prandtl number of steam at atmospheric conditions as functions of temperature (K):

$$\rho_s = 3.4503922053 \cdot 10^{-6} \cdot \mathrm{T_s^2} - 4.2053105100 \cdot 10^{-3} \cdot \mathrm{T_s} + 1.6780736016 \tag{K. 1}$$

$$c_{ps} = 6.5126050420 \cdot 10^{-6} \cdot \mathrm{T_s^2} - 5.9701890756 \cdot 10^{-3} \cdot \mathrm{T_s} + 3.3463666843 \tag{K. 2}$$

$$\mu_s = 1.4604198182 \cdot 10^{-7} \cdot \mathrm{T_S} - 3.4227408243 \cdot 10^{-5} \tag{K. 3}$$

$$k_s = 0.0000837273 \cdot \mathrm{T_s} - 0.0066655591 \tag{K. 4}$$

$$\mathrm{Pr}_s = -2.4181818182 \cdot 10^{-4} \cdot \mathrm{T_s} + 1.0730526364 \tag{K. 5}$$

Density (kg/m^3), specific heat (kJ/kgK), dynamic viscosity (m^2/s), thermal conductivity coefficient (W/mK) and the Prandtl number of water at atmospheric conditions as functions of temperature (K):

$$\rho_w = -3.8388230179 \cdot 10^{-3} \cdot \mathrm{T_w^2} + 2.0464231519 \cdot \mathrm{T_w} + 727.82179247 \tag{K. 6}$$

$$c_{pw} = -0.0007200000 \cdot \mathrm{T_w} + 4.4136680000 \tag{K. 7}$$

$$\begin{aligned}\mu_w = &-2.8193473193 \cdot 10^{-9} \cdot \mathrm{T_w^3} + 2.9071381702 \cdot 10^{-6} \cdot \mathrm{T_w^2} \\ &-1.0039290059 \cdot 10^{-3} \cdot \mathrm{T_w} + 0.11651081841\end{aligned} \tag{K. 8}$$

$$k_w = 0.3765493686706\mathrm{Ln}(\mathrm{T_w}) - 1.5368285999447 \tag{K. 9}$$

$$\mathrm{Pr}_w = 1.7696969697 \cdot 10^{-3} \cdot \mathrm{T}^2 - 1.2346642424 \cdot \mathrm{T_w} + 217.42872345 \tag{K. 10}$$